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A New Method of Analyzing the Cylinder Performance of Reciprocating Engines

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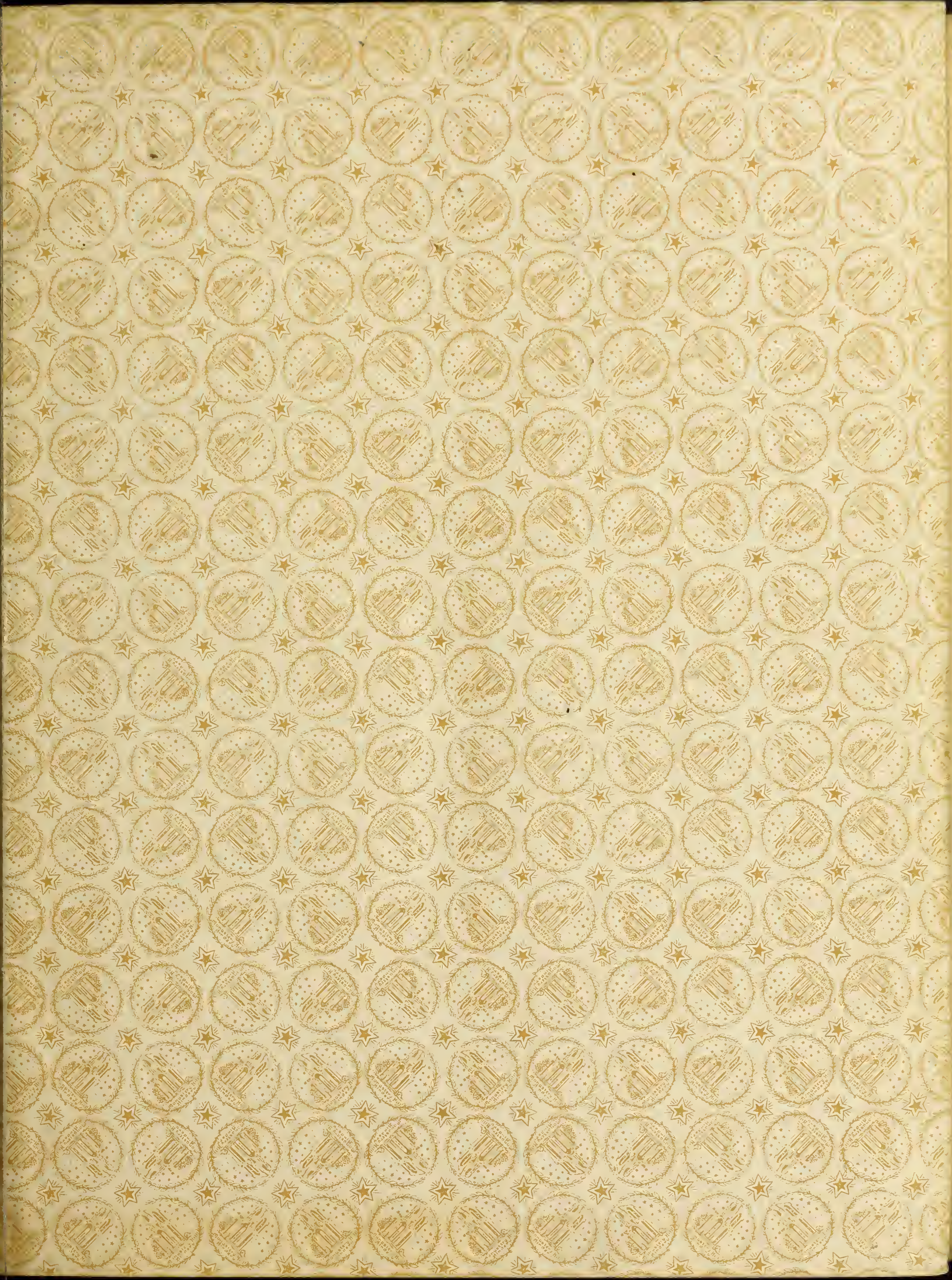

A NEW METHOD OF ANALYZING THE CYLINDER PERFORMANCE OF RECIPROCATING ENGINES

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BY

JEAN PAUL CLAYTON
B. E. TULANE UNIVERSITY OF LOUISIANA, 1909

THESIS

SUBMITTED IN PARTIAL FULFILLMENT OF THE
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IN

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Jean Paul Clayton

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BE ACCEPTED AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE

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CHAPTER I.

INTRODUCTION.

1. PRELIMINARY.

The cylinder performance of a reciprocating engine, using an elastic fluid for the working medium, may be considered from two points of view: its performance as a heat engine, or the efficiency of transformation of the heat available into indicated work; and its performance as a mechanism, or the measure of perfection attained in the distribution of the working fluid and in preventing leakage past the valves, piston, or piston rods. In investigating actual performances, there are various devices and means which aid to a certain degree, in separating the different phases of performance. This fact enables defects to be corrected when they are located.

Most of our knowledge of cylinder performance is obtained from indicator diagrams, taken by means of short indicator connections, accurate instruments, and correct reducing motions. From these diagrams we obtain a measure of work performed in one cycle of operation. Thus the immediate result is, that with a certain amount of heat available, we have obtained a certain proportion of it as work. This proportion is expressed as a thermal or potential efficiency.

The shape of the expansion and compression curves of the diagram vary considerably. Writers have developed hypotheses to account for the variations in the shape of these curves. About two years ago, while constructing assumed gas engine diagrams to be used in a gas engine design, the writer plotted several actual diagrams on logarithmic cross-section paper to find the value of $\underline{n}$ for the curves. A few steam diagrams were also plotted for the same reason. In the latter, for diagrams taken from the same cylinder at different values of cut-

off, it was noted that the value of n for the expansion curve was higher as the cut-off was longer. After considerable thought, it was decided that there might be a relation, in any one cylinder between the value of n for the expansion curve and the quality of the steam mixture at cut-off, as this quality improved when the cut-off was lengthened. One fact that seemed to confirm this belief was that, with superheated steam, when this quality at cut-off is high, the value of n for the expansion curve is much higher than with saturated steam.

This hypothesis seemed reasonable, so a series of tests was planned which would disclose this relation if it existed.

A Corliss engine was selected because of the fact that in this type all the steam used had to pass through the cylinder. Sixty separate tests were made, and the following conditions maintained. With saturated steam, the effect on the value of n of changing the cut-off, the steam pressure, and the speed was observed. This was repeated with steam superheated about 100° F.

These tests showed that there was a very definite relation between the value of n and the quality at cut-off. This fact enabled the writer, with diagrams from this engine, and the relations of n and quality found in the tests, to reverse the process used in finding this relation, and to compute the actual steam consumption to within an error of less than 5% of its weight as measured at the condenser. The importance of finding such a relation will be easily seen.

After long and careful study of the values of n found in various engines using different media, and the variation in this value for the same curve in any particular case, there were found, one by one, other very important facts which related to the detection of leakage, close location of the events of the cycle, and a rational method of obtaining clearance. As a result, a very complete analysis of cyl-

inder performance has been disclosed.

The indicator diagrams to which this analysis applies are those taken from the cylinders of reciprocating engines using any elastic fluid for the working medium and having as a part of the cycle an expansion or compression of the medium.

The development of this new method of analysis, and comparison with the existing methods is found in the following Chapters of this thesis. The plan of presentation will be briefly outlined.

Chap. II contains a very brief summary of the results of the analyses of indicator diagrams that may be considered standard practice at the present time.

Chap. III describes the plotting of logarithmic diagrams, and gives values of $\underline{n}$ for curves found in practice by means of this diagram.

Chap. IV gives the values of $\underline{n}$ for adiabatic change of state of the different media with which this thesis deals.

Chap. V gives the results of laboratory tests of a Corliss engine.

Chap. VI contains the discussion of the exhibit of results, and develops the new method of analysis in detail.

Chap. VII shows applications of the new method to particular cases, and gives suggestions for new fields of application.

Chap. VIII is devoted to the discussion of the existing methods of analysis, as viewed from the standpoint of the new method.

Chap. IX gives the conclusions derived from the results of the Research.

It is believed, after working with the new method, that the existing methods of analysis are in a very crude state, and that many of the deductions made are absolutely without foundation or meaning.

2. ACKNOWLEDGEMENT.

The writer desires especially to thank Professor G. A. Goodenough for his hearty encouragement and appreciation of the subject matter of this thesis since its inception.

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The writer wishes to acknowledge the assistance in running the tests rendered by the following Senior students: Jacobsen, Schuster, Parmely, Hodgson, Janda, Butzer, and Wood, all of the Class of 1910, and Hasberg, Hagedorn, Allen, Herrcke, Cobb, and Ponder, all of the Class of 1911.

He also wishes to thank the various firms who furnished tests for analysis.

CHAPTER II.

REVIEW OF THE EXISTING ANALYSES OF CYLINDER PERFORMANCE FROM INDICATOR DIAGRAMS.

1. THE PV-DIAGRAM.

The indicator diagram, or PV-diagram, is an autographic record of pressure in the cylinder of a reciprocating engine at the different parts of the stroke. This record is made by an indicator, of which many forms are on the market.

The indicator was invented by James Watt and was used by him in improving the steam distribution of his engines. Watt kept the device secret for many years, thereby securing a great advantage over his competitors of that period.

Following the invention by Chas. B. Richards, in 1862, of a multiplying straight-line motion, so that the deflections of the spring were multiplied many times, thereby allowing a heavier and more reliable spring to be used, many high-grade and thoroughly reliable instruments have appeared.

The indicator was at first applied to steam engines only, but later, with the advent of other forms of power, to gas engines, pump cylinders using any liquid, compressed air engines, compressors of air, ammonia, and different gases, and to the barrels of ordnance pieces.

For an interpretation of the different lines of a PV-diagram, the reader is referred to the works of such writers as Thurston, Peabody, Pray, or to handbooks issued by the different indicator manufacturers.

2. THE APPLICATION OF INDICATOR DIAGRAMS IN THE PV FORM.

For details of the processes briefly outlined here, the reader is referred to books dealing with the subject.

a. Indicated Power.

The most common use of the diagram is to obtain a measure of the work developed in the cylinder indicated. For this purpose, the area of the diagram is obtained, and the proper scales are used to reduce this area to work.

b. Valve Setting.

The cyclic events, such as the opening and closing of valves, are shown faithfully on the diagram. By interpreting the meaning and location of such events, the power developed in the opposite ends of a cylinder may be equalized, the valves may be set so as to effect the best economy, and ignition in a gas engine may be made to occur at the proper moment. It is standard practice to set the valves of steam engines, fitted with indicator connections, so that the events of the cycle occur at the proper points as shown by the indicator. These connections are furnished as regular equipment on large engines.

c. Location of Cyclic Events.

It is often useful to locate the positions of such events as lead, cut-off, release, and compression in a steam cylinder, and similar events in the cylinders using other media. These events are generally shown on the diagrams in such a manner as to be located with fair accuracy.

d. Hirn's Analysis of Steam Diagrams.

Hirn's analysis consists in accounting, by means of the pressures and volumes of steam shown by the indicator, for the quantities of heat present in the steam at different points between the events of the beginning of compression and release. Thus the loss of heat to the cylinder walls, due to initial condensation, is revealed, and the restoration of part of this lost heat, during the expansion, is shown. It is the heat balance of one stroke of the engine, and is used in reporting elaborate tests, and in research relating to initial

condensation, re-evaporation, the use of steam jackets, superheated steam, high speeds, and any work where it is desirable to know the heat losses in the cylinder itself.

For a detailed working out of a Hirn's analysis, the reader is referred to the works of Carpenter, Peabody, and Creighton.

e. The Use of Indicator Diagrams in Compression Practice in determining Number of Stages.

When air or other gas is compressed, it is usually desirable to compress it as nearly isothermally as possible, in order to reduce the amount of work to be done upon it. This result is accomplished more or less completely either by water cooling the compressor cylinders, or by dividing the compression in two or more stages and cooling the gas in intercoolers as well as in the cylinders. The simple diagrams and combined diagrams from compressors show to what extent the desired result has been accomplished, and they have been widely used for that purpose.

3. THE CURVES OF THE PV-DIAGRAM.

a. The Expansion Curve.

The character and form of the expansion curves of steam diagrams, when compared with a rectangular hyperbola which is passed through some point on the expansion curve, have been used to draw deductions concerning the tightness of the piston and valves. It has been found, after many trials, that most engines using saturated steam at normal load give diagrams of which the expansion and compression curves approximate an equilateral hyperbola. Some writers, knowing this fact, have maintained therefore that these curves should necessarily be close to an hyperbola, and they regard any curve departing much from it as good evidence of some abnormal influence in the cylinder.

The clearance of steam cylinders can be approximated by as-

suming either the expansion or the compression curve to be an equilateral hyperbola. The correctness of the result is entirely dependent upon how close the curve used approximates an hyperbola.

The expansion curves of gas engine cards reveal the effects of too poor or too rich a mixture. Excessive leakage is also clearly discernable on some curves.

The re-expansion curves of compressors show the presence of leakage (if excessive) of the discharge valves.

b. The Compression Curve.

The compression curve in steam diagrams sometimes shows in a clear manner, by its variation in form, the effects of leakage or excessive condensation. It is not used as much in the hyperbola test as is the expansion curve.

Leakage of pistons or suction valves in compressors is sometimes discernible from the form of the compression curve.

4. DERIVED DIAGRAMS.

There are two forms of diagrams, derived from the PV-diagram, which are of considerable importance.

The more important form is the temperature-entropy diagram. The amounts of work shown on the PV-diagram, as co-ordinate of P and V, are changed to equivalent unit amounts of heat, as co-ordinates of T, absolute temperature, and of S, the entropy measured from 32° F. The areas are then products of T and S, and represent quantities of heat, per unit weight of substance, involved, in the cycle considered.

In the case of saturated steam, we are then enabled to measure the quantity of heat, per unit of weight, supplied to the engine cylinder, and the heat available for work, between the temperature limits of the steam supplied and of the condenser, if the process were adiabatic. Of this heat available, a portion is expended in indicated work, some is lost by wire-drawing, initial condensation, condensation

during expansion, incomplete expansion, clearance, back-pressure, and receiver losses in compound engines only. One of these losses, that by condensation, is partially restored by re-evaporation during the latter part of expansion and during the exhaust. The temperature-entropy diagram is not used extensively in engine testing, and, as a matter of fact, is of no value for obtaining quantitative results by the measurement of the area on the TS-plane, as the actual process is an irreversible one.

The other form of derived diagram is one of the expansion or compression curves plotted on logarithmic cross-section paper from the PV-diagram. This is used to obtain graphically the value of n from these curves. This partial diagram is very rarely used.

5. SUMMARY.

Our knowledge of cylinder performance by means of the indicator may be divided into two classes: first, the satisfactory results; and second, the unsatisfactory results.

a. The following results are of the class that is in a satisfactory state:

1. Indicated Work.
2. Aid in Valve Setting.
3. Heat Balance in Steam Cylinders.
4. Detection of Leakage(if very large)during expansion or compression.

b. The following problems are in the class which is in a very unsatisfactory state:

1. Accurate determination of clearance from all cylinders using elastic media.
2. Close location of cyclic events.
3. Reliable detection of moderate leakage with the engine in motion.

4. Measure of the actual steam consumption from steam diagrams.
5. Separation of initial condensation from leakage in steam cylinders.
6. Division of feed between the two ends of a steam cylinder for the application of Hirn's analysis.

CHAPTER III.

THE PLOTTING OF INDICATOR DIAGRAMS ON LOGARITHMIC CROSS-SECTION PAPER.

1. DESCRIPTION OF LOGARITHMIC CROSS-SECTION PAPER.

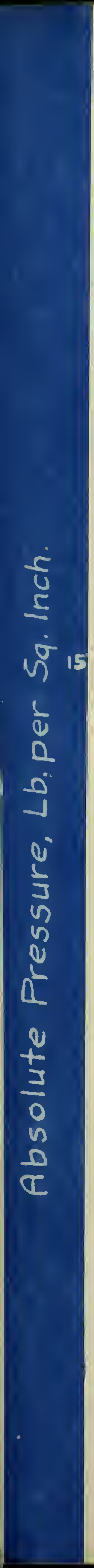
Logarithmic cross-section paper differs from rectangular cross-section paper in that the distances from the origin are proportional to the logarithms of the numbers to be plotted instead of the numbers themselves. This system of coordinates gives an uneven scale similar to that on a slide rule.

The numbers of the divisions are placed opposite the lines corresponding to the logarithms, as on a slide rule, instead of the values of the logarithms of the numbers. This fact aids in plotting, as the logarithms are used without having to ascertain their values.

The logarithmic paper used in this investigation consists of four squares as shown in Fig. 1. These squares are five inches each way, making the four squares together as shown ten inches each way. The use of four squares enables values to be plotted ranging from 0.1 to 10.0, 1.0 to 100.0, etc., thus giving a range of ten times the values obtainable if only one square were used.

2. METHOD OF TRANSFERRING THE PV-DIAGRAM TO LOGARITHMIC CROSS-SECTION PAPER.

The coordinates of the PV-diagram are pressure and stroke, the latter being proportional to the volume displaced by the piston. The coordinates of several points on the PV-diagram must be found in terms of absolute pressures, preferably in pounds per square inch, and absolute volumes, preferably in cubic feet. The scale of units used is not material as long as it starts at the line of zero pressure, or of zero volume. However the units are more easily manipulated afterwards if they are the same as are used in the steam tables.


Absolute Pressure, Lb. per Sq. Inch.
15

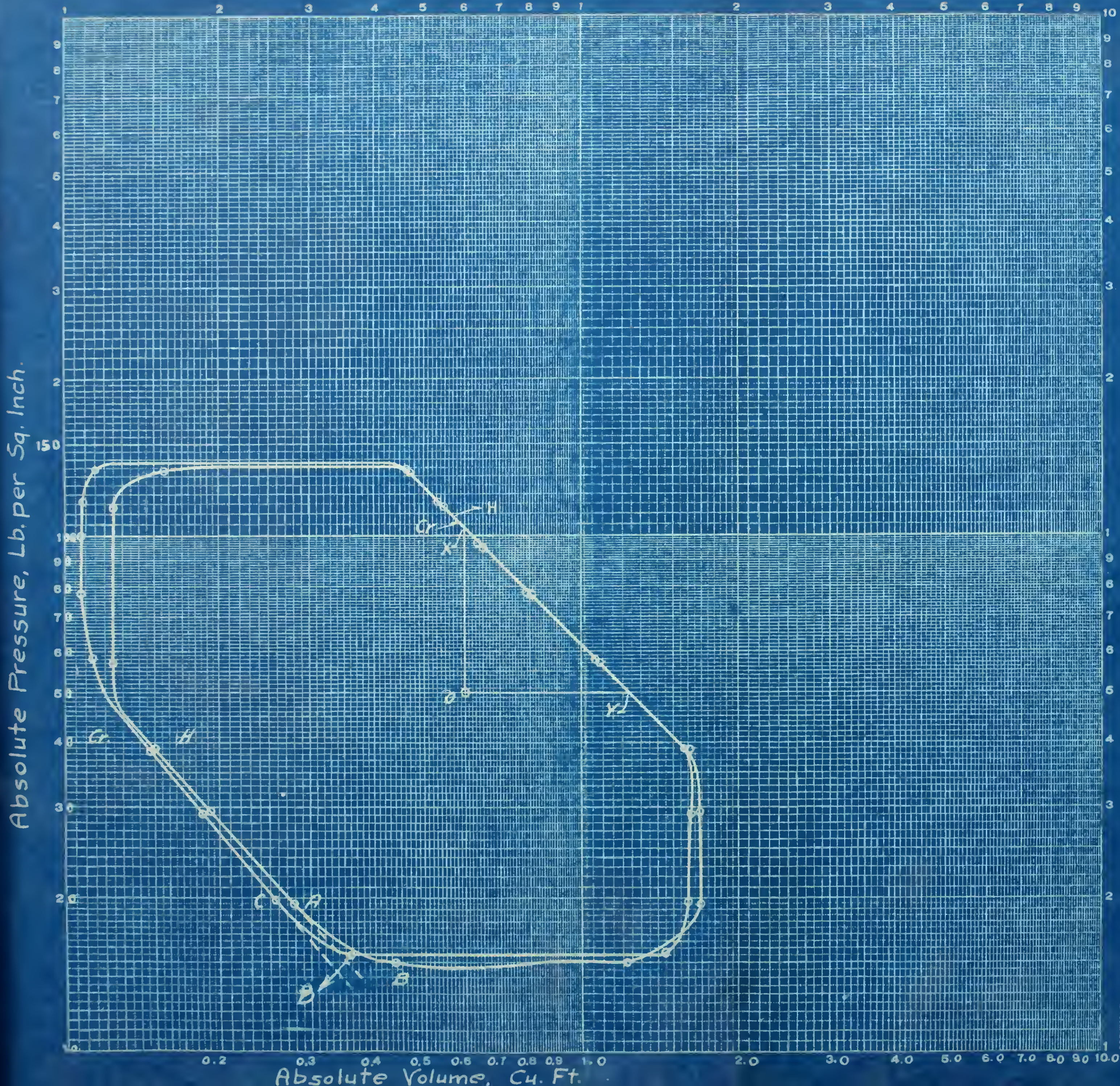


Fig. 1
Logarithmic Diagrams
Plotted from Fig. 2

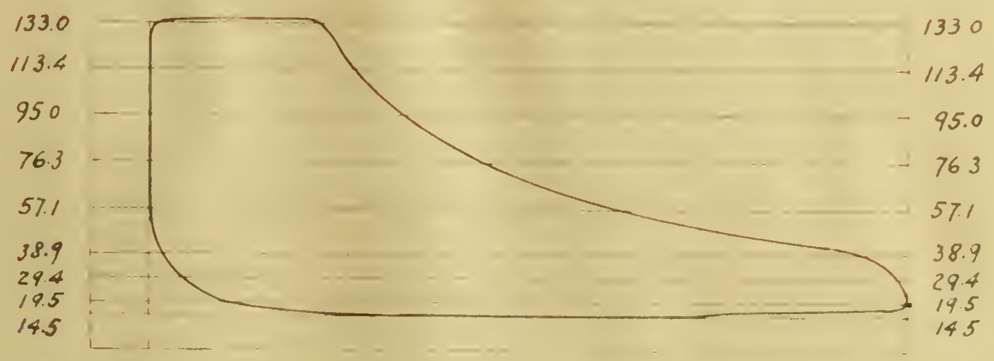


Fig. 2 Head End
Scale - 77.0 #

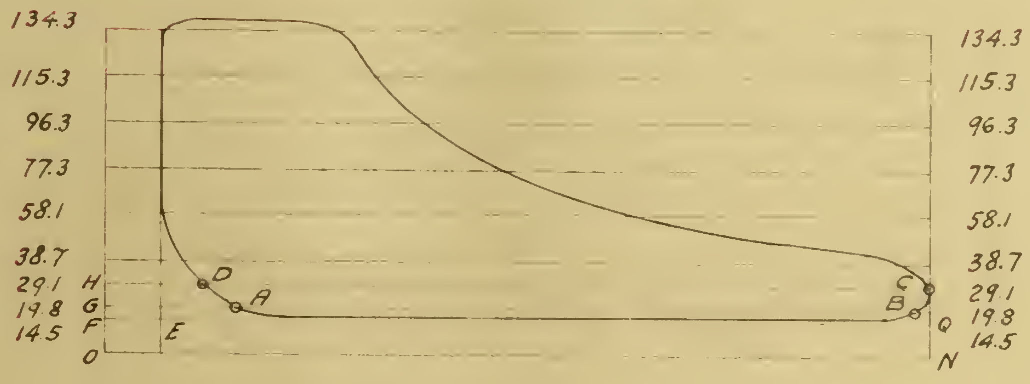


Fig. 2 Crank End
Scale 79.0 #

Fig. 2 Cr shows a typical steam indicator diagram, ABYX, taken from the engine described in Appendix I. Perpendiculars QR and EX are drawn to the atmospheric line EQ, and pass through the extreme stroke positions of the diagram. The distance EQ is then the length of the diagram. OM is laid off perpendicular to the atmospheric line EQ, extended, which was drawn by the indicator. It is the line of zero volume, and is drawn at a distance FE from the admission end of the diagram EX, this distance FE being the same length in per cent of the line EQ, or length of the diagram, as the proportion that the per cent of clearance, or waste space of the cylinder, bears to the piston displacement. If the clearance is 3.2%, then FE is 3.2% of the length EQ. ON is the line of zero pressure and is drawn a distance FO below the atmospheric line, to the scale of the spring used in obtaining the PV-diagram. This distance is proportional to the barometer reading, corrected for temperature, prevailing at the day and place of the engine test. From ON points are laid off on QR and EX corresponding to the absolute pressures, at the intervals where it is desired to read the corresponding volume.

Fine lines are drawn connecting similar pressure points. These are shown as 19.8-19.8, 29.1-29.1, etc. The volumes, such as GA, GB, HD, HC, etc. are read off in hundredths of an inch and multiplied by the constant ratio which one inch of length on the diagram bears to the displacement of the piston. Thus one inch of length on the diagram may be equivalent to 3 cubic feet of piston displacement. This process is continued at intervals until the coordinates of from ten to thirty points are determined. These coordinates of P and V, are then plotted as shown in Fig. 1, which is the logarithmic diagram derived from the PV-diagram in Fig. 2. A smooth curve is drawn through the points thus obtained, and the diagram is in shape to be studied.

The facts obtained from this form of derived diagram are the

basis of the new method described in the following parts of this thesis. The writer proposes for this figure the name, logarithmic diagram, and this name will be used throughout this investigation.

3. MATHEMATICAL RELATIONS EXISTING BETWEEN THE PV-DIAGRAM AND THE LOGARITHMIC DIAGRAM.

a. Expansion and Compression Curves.

It has been found, under the conditions where the amount of steam or gas remains constant, and without abnormal influences, such as leakage, that expansion and compression take place substantially according to the law $PV^n = C$. This equation when plotted on rectangular cross-section paper gives a curve depending in its shape on the values of P, V, n, and C. When this curve is plotted on logarithmic paper, it becomes a straight line depending for its slope on the value of n. The relations for such curves follow:

Given $PV^n = C.$

Taking the logarithm of both terms,

$$\log P + \underline{n} \log V = \log C.$$

Transposing, $\log P = - \underline{n} \log V + \log C.$

This is of the form of the straight line

$$y = mx + b$$

where $y = \log P$

$m = - n$

$x = \log V$

$b = \log C$

Thus $m = - n$, the slope, or measure of inclination of the line to the axis log V. In Fig. 1, for example, at a point X below the line $\log P = - \underline{n} \log V + \log C$, draw OX parallel to the axis, log P, and OY parallel to the axis, log V. The slope will be the value of the ratio $OX/-OY = - \underline{n}$. OY is negative, being measured to the left, giving n its negative sign.

4. VALUES OF n FOR EXPANSION AND COMPRESSION CURVES FROM PRACTICE.

a. Steam Engines.

The values of n for the expansion and compression curves of indicator diagrams are subject to a wide variation. It has been found however, in the great majority of cases examined, that the expansion and compression curves are practically exact polytropic curves of the form $PV^n = C$.

Zeuner gives the values of n found by several early investigators. These values were mostly taken from the diagrams of small and slow-speed engines. Leloutre found that the value of n was practically constant for any one case, but varies greatly in different engines, according to the initial pressure and ratio of expansion. Lüders found values ranging from 0.903 to 0.535. Zeuner found values of from 0.900 to 0.436 from diagrams taken by Hallauer from a Corliss engine. In none of these cases was n found to be as high as 1.0. Zeuner concludes that the value is generally close to 1.0, and does not vary much either way.

Heck shows diagrams from the cylinders of compound Corliss engines, locomotives, pumping engines, and marine engines, all using saturated steam. The values of n for the expansion curves of the H.P. cylinders of these engines are all close to 1.0. The values for the L.P. expansion curves are generally less than 1.0, ranging from about 0.95 to 0.90. The values for the compression curves shown do not depart far from 1.0.

The writer has examined a large number of curves by the graphical method described in Chap. II. These examples, which are given in Tables 1-8, may be classed as follows:

Class A - Corliss, Four Valve, and Riding Cut-off Types, all simple expansion.
B - Same, Compound.

TABLE I

VALUES OF n FROM CLASS A STEAM ENGINES

Size-inches	Value of n		Number of Diagrams Examined	Make	Remarks
	Expansion	Compression			
$18\frac{1}{4} \times 45$	0.952	1.007	2	Greene	Saturated Steam
"	0.998	1.115	2	"	
$15\frac{1}{4} \times 24$	1.049	1.170	2	Buckeye	
$26\frac{1}{2} \times 36$	0.928	1.063	2	"	
"	0.823	1.036	2	"	
"	0.996	1.121	2	"	
20×30	0.624	0.985	2	"	
"	1.024	0.992	2	"	
18×42	1.047	0.944	2	Unknown	
16×32	1.065	1.154	2	"	
"	1.111	1.341	2	"	
$16\frac{5}{8} \times 32$	0.964		2	"	
$26\frac{3}{16} \times 48$	1.051		2	"	
17×24.2	1.131		4	"	
23×60	1.098		2	"	
$28\frac{1}{2} \times 59\frac{1}{2}$	1.108		2	"	
11 –	Engines	Total	34 –	Diagrams	

TABLE 2
VALUE OF n FROM CLASS B STEAM ENGINES

Size - Inches	Expansion		Compression		Number of Diagrams Examined	Make	Remarks
	H P	L.P	H P	L P			
$18\frac{3}{4} \times 36 \times 36$	1.060	0.896			4	Buckeye	Saturated Steam
$15 \times 40\frac{1}{2} \times 27$	0.955				2	Fleming	
20 x 36 x 48	1.055	0.973			4	Watts Campbell	
21 x 42 x 36	1.068		0.970	0.742	4	Gaskell	
25 x 50 x 37	0.841	0.879	0.937		8	Worthington	
22 x 40 x 60	1.070	1.009	0.850		4	Harris-Corliss	Cyls. Jacketed
20 x 30 x 48	1.079	0.977	1.018	1.272	4	Watts Campbell	Unjacketed
10 x 40 x 48	1.090	0.950			4	Cooper Corliss	
	1.048	1.116			4		Cyls Jacketed
8 Engines				Total	38	Diagrams	

TABLE 5
VALUE OF n FROM CLASS E STEAM TABLES

TABLE 3
VALUE OF n FROM CLASS C STEAM ENGINES

Size - Inches	Expansion			Number of Diagrams Examined	Remarks
	H.P.	I.P.	L.P.		
23 x 54 x 80 x 60	0.987	1.112	0.984	6	Saturated Steam Cylinders Jacketed
	Compression				
	0.964	0.416	0.972		
1 Engine	Total			6	Diagrams

TABLE 4
VALUES OF n FROM CLASS D STEAM ENGINES

Size - Inches	Expansion		Number of Diagrams Examined	Make	Remarks
	H.P.	L.P.			
28 x 58 x 48	1 046	1.070	2	Mc Intosh	Superheated Steam
23 x 48 x 48	0.955	0 969	2	Seymour & Co	Jackets not used
		1.108	1		
	0.905	0.925	2		
	1.119	1.075	2		
29 x 60 x 56		1.231	1		
29 x 60 x 56		1.172	1		
18 x 38 x 42	1.170	0.925	2		
18 x 38 x 42	1.118	1.087	2		
31 x 64 x 48	1.024	0.973	2		
7 Engines	Total	17	Diagrams		

TABLE 6

VALUES OF $\underline{n}$ FROM CLASS F STEAM ENGINES

Size - Inches	Expansion H.P.	Number of Diagrams Examined	Make	Remarks
$14\frac{1}{2} \times 13$	0.970	2	Unknown	Saturated Steam
16×16	1.027	2	Ide	
$11\frac{1}{2} \times 12\frac{1}{2}$	0.706	2	Unknown	
3 -	Engines 6 Total Diagrams			

TABLE 7

VALUE OF $\underline{n}$ FROM CLASS G STEAM ENGINES

Size - Inches	Expansion		Number of Diagrams Examined	Make	Remarks
	HP	L.P.			
$13 \times 22 \times 13$	1.073	0.874	2	Westinghouse	Saturated Steam
	1.054	0.863	2		
	1.062		1		
1 Engine	Total		5 - Diagrams		

TABLE 8

VALUES OF n FROM CLASS H STEAM ENGINES

Size-Inches	Expansion	Compression	Number of Diagrams Examined	Make	Remarks
16 x 24	1 003		4	Schenectady #2	Saturated Steam
22 x 30	0 985	0 987	2	I. C. #94	
	0.981		2		
22 x 30	0.975	0.970	2	I. C #920	
	1.006	1 125	2		
16 x 24	0.974	1.208	2	Schenectady #3	Superheated Steam
	1.124	1 179	2		
	1 167	1.195	2		
	1 046	1.188	2		
	1 149		2		
3 Engines	Total 22 Diagrams				

TABLE 10

VALUES OF n FROM COMPRESSED AIR LOCOMOTIVES

Size Inches	Expansion		Number of Diagrams Examined	Make
	H.P	L.P		
5 x 10 x 10	1.123	1.354	4	Porter
6 x 10	1 369		4	
2 Engines	Total 8 Diagrams			

C - Same, Triple expansion.

D - Gridiron Valve Type, Compound.

E - Poppet Valve Type.

F - High Speed engines, single valve types.

G - Single Acting engines.

H - Locomotive engines.

The classes cover the values derived from 138 diagrams taken from 36 separate engines. In Chap. V are given the values obtained from 60 sets of diagrams taken from a 12 in. X 24 in. Corliss engine.

The values given in Tables 1-8 are the average values from both ends of one cylinder, and, in the case of two-cylinder simple engines, from both ends of both cylinders. In many cases only the expansion curves were examined.

These diagrams were taken from cylinders ranging in size from 11 1/2 in. X 13 in. up to 80 in. X 60 in., and in speed from 300 R.P.M. down to 23.6 R.P.M. Most of the cylinders were unjacketed. Examples of both saturated and superheated steam are shown. It will be seen, therefore, that the range of the exhibit is very broad.

It is evident that the values of *n* are subject to extremely wide variations, the range being from 0.436, found by Zeuner up to 1.341, found by the writer, in the case of an engine using highly superheated steam. It is true, however, that the average of all the values cited is not far from 1.0.

b. Gas engines.

The values of *n* for the curves of gas engine diagrams have a smaller range of variation than that found in steam diagrams.

Güldner finds that the value of *n* for the compression curves varies from 1.30 to 1.38, with an average of about 1.35, but he mentions rare values higher than the adiabatic value, due to high temperature of cylinder walls, and the consequent addition of heat to the

gas during compression. He finds that n for expansion varies normally from 1.35 to 1.50, but cites lower values than 1.35 due to leakage, and higher values than 1.50 due to excessive temperature of cylinder walls from poor cooling.

Wimperis gives the values found by Professor Burstall from diagrams taken during 10 tests on the same engine, presumably. Professor Burstall finds n for expansion to vary from 1.199 to 1.344, with an average of 1.288. The values of n for compression vary from 1.345 to 1.364, with an average of 1.352.

The writer, after examining 17 diagrams from 5 separate engines, finds values for expansion and compression as given in Table 9. These values given show that the variation of n for expansion is ordinarily from 1.10 to 1.37, while for compression it is from about 1.20 to 1.40, leaving out of account very high values due to imperfect cooling and very low values due to leakage and to cool walls during the period of "starting up". The values of n, for expansion in large cylinders with lean gases, are low, being about the same as those found in steam cylinders using superheated steam.

c. Compressed Air engines.

Very little data has been found concerning the values of n from diagrams taken from compressed air engines. Locomotives comprise the larger part of this class which use air expansively.

The writer examined the curves of 8 diagrams taken from the four cylinders of two locomotives, one a compound, and the other a simple expansion type. These are given in Table 10.

The expansion curves of these diagrams were very satisfactory, but the compression curves were all irregular, and no values were obtained from most of them. This irregularity was probably due to the vibration that is present in most locomotives under test conditions.

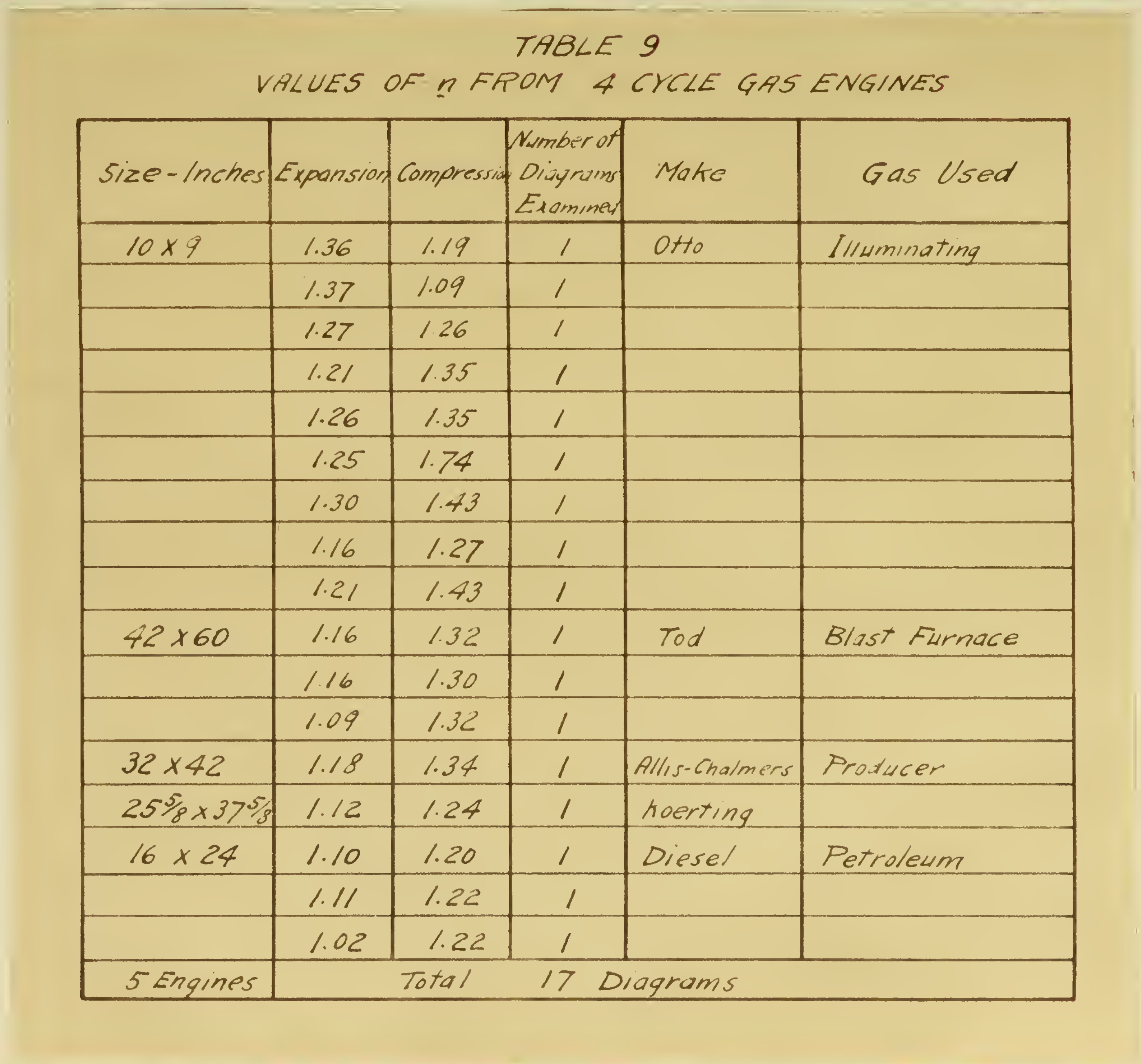


TABLE 9

VALUES OF η FROM 4 CYCLE GAS ENGINES

Size - Inches	Expansion	Compressio	Number of Diagrams Examined	Make	Gas Used
10 x 9	1.36	1.19	1	Otto	Illuminating
	1.37	1.09	1		
	1.27	1.26	1		
	1.21	1.35	1		
	1.26	1.35	1		
	1.25	1.74	1		
	1.30	1.43	1		
	1.16	1.27	1		
	1.21	1.43	1		
42 x 60	1.16	1.32	1	Tod	Blast Furnace
	1.16	1.30	1		
	1.09	1.32	1		
32 x 42	1.18	1.34	1	Allis-Chalmers	Producer
$25\frac{5}{8} \times 37\frac{5}{8}$	1.12	1.24	1	Koerting	
16 x 24	1.10	1.20	1	Diesel	Petroleum
	1.11	1.22	1		
	1.02	1.22	1		
5 Engines		Total	17 Diagrams		

TABLE 11

VALUE OF n FROM AIR COMPRESSORS

Size-Inches	Compression		Number of Diagrams Examined	Make	Remarks
	H.P.	L.P.			
12¼ x 18¼ x 12	1.336	1.219	4	Ingersoll-Seargent	Small amt. of cooling water
	1.266	1.254	4		Large " " "
1 Compressor		Total	8	- Diagrams	

TABLE 12

VALUE OF n FROM AMMONIA COMPRESSORS

Size-Inches	Compression	Number of Diagrams Examined	Make	Remarks
12½ x 18	1.148	1	York	Dry Comp Sing Acting
12½ x 18	1.186	2		" " Doub "
	1.235	2		Wet Compression
2 Compressors	Total	5	Diagrams	

TABLE 13

VALUE OF n FROM GAS COMPRESSORS

Size-Inches	Compression	Number of Diagrams Examined	Make	Remarks
28¼ x 24	1.140	2	Ingersoll-Seargent	Illuminating gas
	1.145	2		
1 Compressor	Total	4	Diagrams	

The values for expansion range from about 1.12 to 1.37.

d. Air compressors.

Only 8 diagrams from one two-stage compressor were available for examination. The values of n for compression are given in Table 11. No values for re-expansion were obtained.

The values for compression in air compressors never vary much, and will generally fall between 1.20 and 1.35.

e. Ammonia compressors.

The examples available of this type of compressor were limited to 5 diagrams from 2 compressors. The compression curves were very satisfactory, but the re-expansion curves were quite irregular. These values will be found in Table 12.

The values found fall between about 1.15 and 1.24, very little variation being observed.

f. Gas compressors.

This class refers to compressors used to raise the pressure of illuminating gas in order to send it to distant points in small pipes. The analysis of the gas compressed in the single case examined is given in Chap. IV. Values of n were obtained for the compression curves only. These values are given in Table 13.

Although taken at different conditions of the speed and the discharge pressure, these values show substantial agreement, but are considerably lower than the values obtained for air compression.

CHAPTER IV.

THEORETICAL VALUES OF n FOR ADIABATIC EXPANSION OF STEAM AND GASES.

1. VALUES FOR STEAM.

a. Saturated Steam, variation of n with initial pressure and quality.

Rankine seems to have been the first to obtain an expression for the adiabatic expansion of steam. He stated as a result of his calculations that the adiabatic change of state may be closely represented by the equation

$$P_1 V_1^n = P_2 V_2^n = C$$

where n = 10/9, or 1.111. Rankine apparently only examined cases where the steam was initially dry at the beginning of expansion.

Grashof concluded that n was more nearly equal to 1.140 for adiabatic expansion of initially dry saturated steam, this being the only condition he examined.

Zeuner in reviewing Rankine's and Grashof's values observed that neither of these writers had examined the case where the steam was initially composed of a mixture of vapor and water, a condition occurring often in engineering practice, and showed that for such a case these values do not hold. He also showed that n varied in value with the initial pressure. Zeuner therefore computed values with the aid of his steam tables for different initial states of pressure and quality. He found that the change in initial pressure had only a relatively small influence when compared with the initial quality and gives the values shown in Table 14 as closely applicable to all pressures common to engineering practice.

He concluded that these values were closely expressed by the equation

TABLE 14

ADIABATIC VALUES OF n FROM ZEUNER

Initial Quality before Expansion x	Calculated Values of n	Values from Equation
1.00	1.135	1.135
0.90	1.125	1.125
0.80	1.115	1.115
0.70	1.103	1.105

TABLE 15

ADIABATIC VALUES OF n FOR VARIOUS INITIAL PRESSURES AND QUALITIES

	Absolute pressure Lb. per sq. inch			
Quality	50	100	150	200
1.00	1.127	1.136	1.140	1.142
0.90	1.123	1.128	1.127	1.129
0.80	1.121	1.115	1.114	1.114
0.70	1.106	1.108	1.102	1.100
0.60	1.096	1.093	1.089	1.084

TABLE 16

ERRORS OF EQUATION

$$n = 1.059 - 0.000315P + (0.0706 + 0.000376P)x$$

	Absolute pressure Lb. per sq inch			
Quality	50	100	150	200
1.00	+0.46 %	+0.09 %	−0.09 %	± 0.00 %
0.90	± 0.00	− 0.18	− 0.09	− 0.18
0.80	− 0.64	± 0.00	− 0.09	− 0.09
0.70	− 0.09	− 0.36	− 0.09	− 0.18
0.60	± 0.00	± 0.00	− 0.09	± 0.00

$$\underline{n} = 1.035 + 0.100\, x_1$$

where x_1 is within the limits of 1.00 and 0.70.

Mr. E. H. Stone, under the direction of Professor Goodenough, computed separately values of $\underline{n}$ for the conditions shown in Table 15 and by the method outlined below, given in his own words.

"Dry saturated steam, or steam of a given quality $\underline{x}$, was assumed to expand adiabatically. For various lower pressures the volumes and qualities were calculated (by the use of the following characteristic equations connecting T, V, x, and $\underline{s}$)....

$$s_1' + x_1 r_1/T_1 = s' + xr/T$$

$$V = x(V'' - V') + V'$$

$$W = u_1 - u = J\left[(q_1 + x_1 \rho_1) - (q + x\rho)\right]$$

...... After the calculations were completed, the simultaneous values of pressure and volume were plotted on logarithmic cross-section paper (described in Chap. III)...... The points were found to lie very nearly in straight lines, thus showing that an exponential equation was applicable. By measuring the slopes of these lines for different initial pressures and qualities, values of $\underline{n}$ for the different conditions were obtained. A separate equation for the values of $\underline{n}$ for each initial pressure was determined by plotting, on coordinate paper, the values of $\underline{n}$ and the initial quality, and found to be of the form

$$\underline{n} = a + bx$$

...... The next step was to combine these into a general equation approximating those of the individual pressures."

This equation was worked out by Prof. Goodenough and given the form

$$\underline{n} = 1.059 - .000315\, P + (.0706 + .000376\, P)\, x$$

It expresses the values found separately as shown in Table 15 so that the errors in % made by its use are as shown in Table 16.

The average error resulting from the use of this equation

from the separately determined values is .18% and the maximum error .64%. By the use of this equation the writer constructed the table of values given in Table 17. In this table the equation was assumed to hold between the pressure limits of 20 to 240 lb. per sq. in. absolute, and the quality limits of 1.00 and 0.50.

In comparing these values of Table 17 with the values given by Zeuner's equation in Table 14, we see that if we compare Zeuner's values with the values computed for 80 lb. pressure that we have a difference of only .1%, but that this is only true at this pressure. The value 1.115 at a quality of 0.80 given by Zeuner holds between the pressures of 20 to 40 lb. and is not in error more than .27% at any pressure between the limits of 20 to 240 lb. If we compare the value of n for saturated steam given by Zeuner, 1.135, with the value of Table 17 at 240 lb., Zeuner's value is small by about .91%. This is the largest discrepancy.

Stone's values given in Table 15 were made under the assumption that the steam expanded to 5 lb. absolute in the case of the pressures of 50, 100, and 150 lb., and to 10 lb. absolute in the case of the pressure of 200 lb.

Returning now to Rankine's value and comparing it with the values of Table 17, we see that for steam initially dry it is low at all pressures. At 20 lb. it is low 1.8%, and at 240 lb. it is low 3.1%. For all pressures and for initial qualities of 0.80 and 0.75 it does not differ more than .36%. These are the only conditions under which the value 10/9 could be used without serious error.

Comparing the values of Table 17 with Grashof's value, 1.140, we see that this value is in much better accord with the conditions of the initial state of dry saturated steam and pressures common to engineering practice. Between the pressure limits of 80 to 240 lb., it does not differ more than .55%, and it is as good an average value as

TABLE 17
ADIABATIC VALUES* OF n FOR VARIOUS INITIAL STATES OF PRESSURE AND QUALITY

	Initial Pressure Lb. per square inch absolute											
Initial Quality	20	40	60	80	100	120	140	160	180	200	220	240
1.00	1.131	1.132	1.133	1.134	1.136	1.137	1.138	1.139	1.141	1.142	1.143	1.145
0.95	1.127	1.128	1.128	1.129	1.131	1.131	1.133	1.132	1.134	1.135	1.135	1.137
0.90	1.123	1.123	1.124	1.124	1.126	1.125	1.127	1.126	1.127	1.127	1.128	1.129
0.85	1.119	1.119	1.119	1.119	1.120	1.120	1.119	1.119	1.120	1.120	1.120	1.121
0.80	1.115	1.115	1.114	1.114	1.114	1.114	1.113	1.113	1.113	1.113	1.112	1.112
0.75	1.111	1.110	1.110	1.109	1.109	1.108	1.107	1.106	1.106	1.105	1.104	1.104
0.70	1.108	1.106	1.105	1.104	1.104	1.102	1.101	1.100	1.099	1.098	1.097	1.096
0.65	1.104	1.102	1.101	1.099	1.098	1.096	1.095	1.093	1.092	1.091	1.089	1.088
0.60	1.100	1.098	1.096	1.094	1.093	1.091	1.089	1.087	1.085	1.084	1.081	1.080
0.55	1.096	1.093	1.092	1.089	1.087	1.085	1.083	1.080	1.078	1.076	1.074	1.072
0.50	1.092	1.089	1.087	1.084	1.082	1.079	1.077	1.074	1.071	1.069	1.066	1.064

*Values Calculated by equation $n = 1.059 - 0.000315P + (0.0706 + 0.000376P)x$.

can be obtained for these conditions.

If we desire a simple equation of the form given by Zeuner which will represent without serious error the values of n for common pressures and qualities, the values found in the column of 140 lb. seem to be the best that can be obtained. Substituting the value of P = 140, in Professor Goodenough's equation we obtain

$$n = 1.015 + .123\ x$$

The errors made by the use of this equation vary from a maximum value of .64% at the quality 1.00, to 1.4% at the quality 0.50.

b. Superheated Steam.

The values of n for superheated steam are considerably higher than the valuesfor saturated steam. This difference may be expected as superheated steam behaves like a gas in its changes of pressure, volume, and temperature. The value of n is not as high, however, as that of most gases, such as air.

Zeuner gives as a result of experimental data and theoretical considerations, a value of n equal to 1.333 or 4/3. He assumes that this value holds for all pressures and temperatures common to engineering practice.

Mr. W. G. Hiller under the direction of Prof. Goodenough, computed separately values of n under the conditions shown in Table 18, using the same method which Stone used for saturated steam. He computed resulting volumes, pressures and temperatures from the following characteristic equations due to Professor Goodenough.

$$\log P = C - K\left(p + \frac{a}{2}p^2\right)$$

where

$$C = \frac{(0.856571 \log T + 0.0001\ T - s - 0.4300)}{0.25392}$$

$$K = \frac{A\ m\ n}{0.25392\ T^{n+1}}$$

$$\log m = 9.50308$$

TABLE 18

ADIABATIC VALUES OF n FOR SUPERHEAT STEAM

	Temperatures						Pressure Lb per sq. inch Absolute		Temperature Degrees of Superheat Deg. F		Value of Range of	
	Degrees Abs		Degrees F		of Saturation							
Entropy	T_1	T_2	T_1	T_2	T_1	T_2	P_1	P_2	°F_1	°F_2	n	Temp Deg. F
1.6000	1100	1000	640.4	540.4	447.1	410.0	409.50	276.30	193.3	130.4	1.301	100
1.6571	1200	800	740.4	340.4	442.3	302.1	390.05	69.06	298.1	38.3	1.303	400
1.6726	1300	800	840.4	340.4	465.2	293.6	490.10	60.83	375.2	46.8	1.300	500
.7046	1300	750	840.4	290.4	439.5	260.3	378.70	35.60	400.9	30.1	1.293	550
.7245	1300	750	840.4	290.4	421.5	250.5	313.30	30.10	418.9	39.9	1.294	550
1.7481	1300	800	840.4	340.4	404.9	254.4	261.00	32.18	435.5	86.0	1.298	500
1.8051	1500	750	1040.4	290.4	420.5	213.0	310.00	15.00	619.9	77.4	1.283	750
1.8180	1500	750	1040.4	290.4	337.1	287.1	113.50	55.00	703.3	3.3	1.293	750

$$n = 3.5$$

$$A = 1/J = 1/777.7$$

$$T = t + 459.6^\circ$$

These values of n were computed without special regard for the initial pressure which varied from 113.5 lb. to 490.1 lb. These values of n were plotted by the writer, as shown in Fig. 3, to degrees of superheat.

A study of the positions of these points seems to show that variation of pressure does not enter to any appreciable extent in influencing the values. The positions of the points lead to the belief that the determining factor is the degree of superheat. The line shown has been taken to represent as closely as the data warrants the relations existing between the value of n and the degree of superheat. For this relation the following equation has been adopted to represent with fair accuracy the values obtained by Hiller.

$$n = 1.312 - 0.000039\ S^\circ$$

where S = superheat in °F.

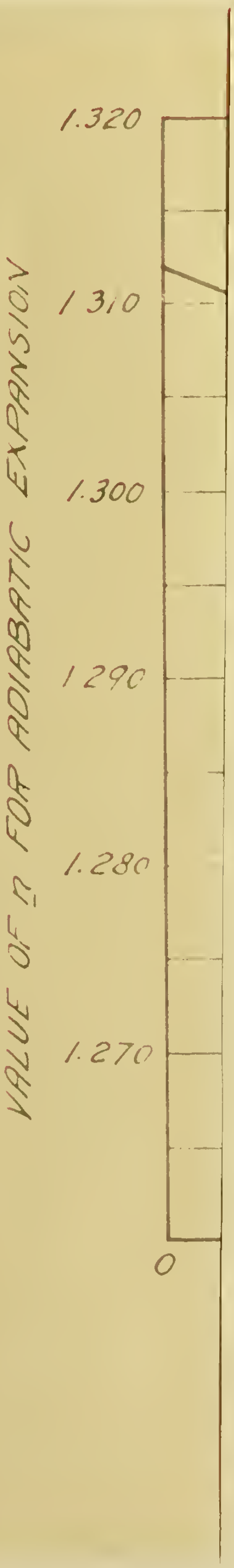
This equation entails an average error of .27% from Hiller's values and a maximum error of .46%. Table 19 was constructed by the use of this equation.

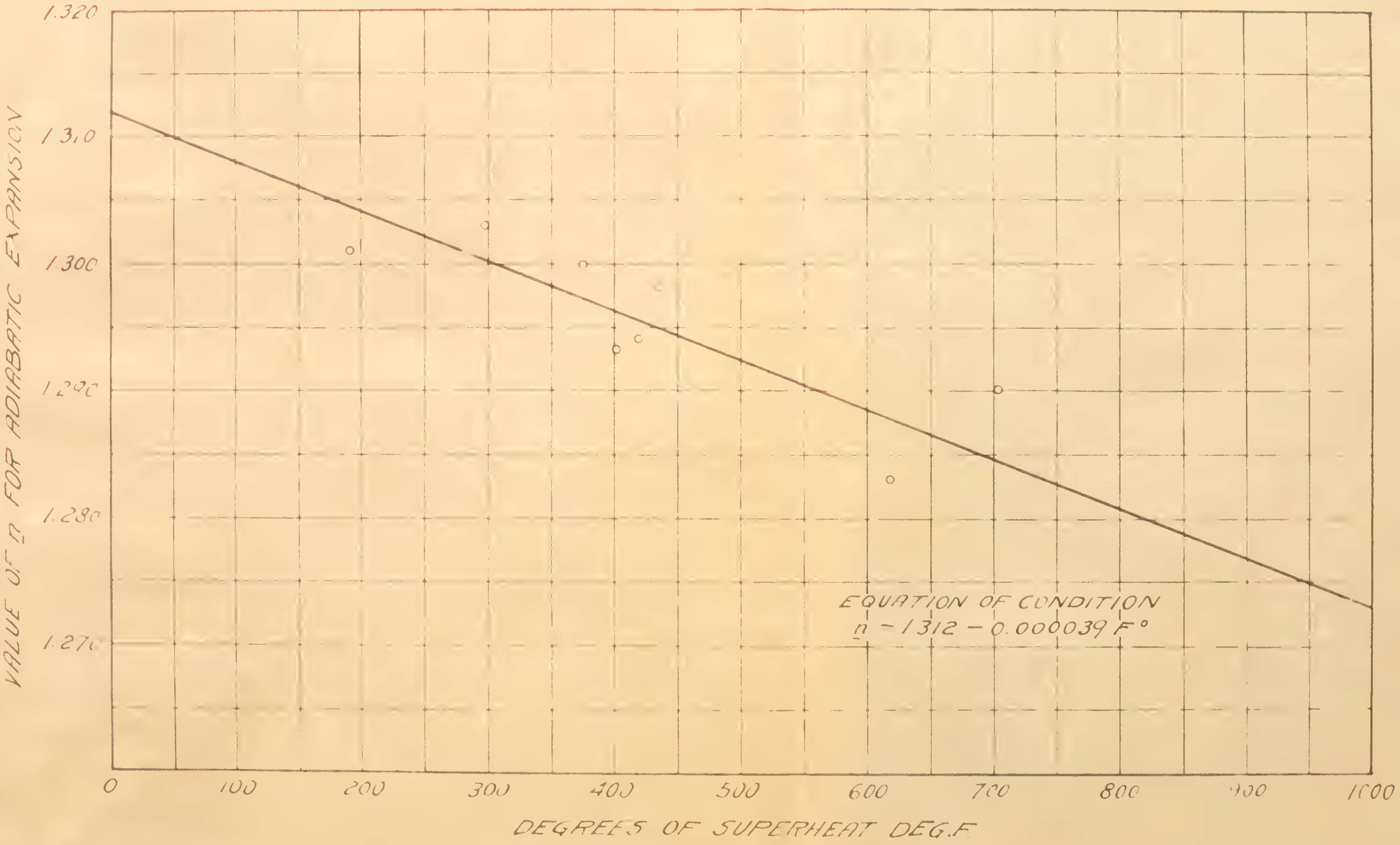
For problems common to the range of engineering practice, the constant value of n, 1.305, could be used without serious error between the limits of superheat of 50° to 200°.

Comparing this value, 1.305, with Zeuner's value, 1.333, we see that under the conditions of which Hiller computed these values, Zeuner's value is high by 2.5%, a considerable difference. Zeuner's own description of the method of obtaining this value shows that it may be in error by at least that amount.

2. VALUES FOR GASES.

The values of n for gaseous mixtures commonly used in gas


VALUE OF n FOR ADIABATIC EXPANSION
1.320
1.310
1.300
1.290
1.280
1.270
0



EQUATION OF CONDITION
n – 1312 – 0.000039 F°

FIG 3
RELATION OF VALUES OF n AND DEGREE OF SUPERHEAT
FOR SUPERHEATED STEAM

TABLE 19

ADIABATIC VALUES OF n FROM SUPERHEATED STEAM FROM EQUATION n [illegible] .000039 F° PRESSURES BETWEEN 100 [illegible]

Degrees Superheat	Value of n
50	[illegible]310
100	[illegible]08
150	[illegible]306
200	1.304
250	1.302
300	1.300
400	1.29[illegible]
500	1.292
600	1.289
700	1.285
800	1.281
900	1.277
1000	1.273

TABLE 20

ADIABATIC VALUES OF n FOR VARIOUS GASES

Gas	Value of n
[illegible]	1.412
[illegible]H_4	1.270
[illegible]O	1.408
[illegible]H_4	1.210
CO_2	1.293
O	1.418
N	1.408
H_2O	1.305
Air	1.410

TABLE 25

ADIABATIC VALUE OF n FOR AN ILLUMINATING GAS

I Constituents	II Percent Weight	III Wt. of 1 Cu. ft., Lb.	IV Wt. in 1 Cu. ft. of the gas, Lb.	V Ratio $\frac{k_p}{k_v}$	VI IV x V
CO_2	2.40	0.12267	0.00295	1.293	0.00381
C_2H_4	9.00	0.07809	0.00703	1.210	0.00851
O	0.50	0.08921	0.00045	1.418	0.000635
CO	31.00	0.07807	0.02420	1.408	0.0341
H	35.20	0.00559	0.00197	1.412	0.00278
CH_4	18.50	0.04464	0.00827	1.270	0.01050
N	3.40	0.07831	0.00266	1.408	0.00375
Total			0.04753		0.06409

$$n = \frac{0.06409}{0.04753} = 1.349$$

engines exhibit considerable variation in the relative proportion of the constituents. The value of n for any gas is the value of the ratio C_p/C_v, and its constancy depends on the relative constancy of the value of C_p and C_v at different temperatures. It seems certain, from the experiments of Mallard and Le Chatelier, that the values of C_p and C_v increase with increase of temperatures, but the ratio C_p/C_v, at temperatures common to actual gas engine cycles, is very closely constant. Most of the following data on the value of n for gases has been taken from Güldner.

The value of n depends, in any particular mixture, on the relative proportion of the constituents. This may be seen from the values of n for different gases given in Table 20, taken from Güldner.

This table shows that by different proportions of these gases in a mixture we may obtain values of n between the extreme limits of 1.210 and 1.418. Thus n for illuminating gas alone, for example, may have a value of 1.332, while for a mixture of this gas with air in the proportion of 1 to 12, the value rises to 1.402, or very nearly the value of pure air. After combustion, the analysis of the gas changes, causing a change of n, the effect being a lowering of its value because of an increase in proportion of CO_2 and H_2O. Thus for the mixture cited above, the value of n after combustion is 1.387, a drop of 1.1%. This, in a hypothetical case of an ideal gas engine cycle, would cause an appreciable difference in the shape of the expansion curve.

Table 21 gives all the data necessary to compute the value of n for the case of one sample of illuminating gas without air. The case of this same illuminating gas with varying ratios of air/gas is given in Table 22. This table gives the data for the same gas with varying combustible ratios of air/gas such as would obtain in gas engine practice.

TABLE 21
ADIABATIC VALUE OF $\underline{n}$ FOR AN AVERAGE GERMAN ILLUMINATING GAS

Gases	Composition by		C_p	C_v	$G \times C_p$	$G \times C_v$
	Volume	Weight				
H	0.485	0.0845	3.430	2.430	0.2890	0 2050
CH_4	0.350	0 4855	0.593	0.468	0 2880	0 2270
CO	0.070	0.1703	0.245	0 174	0 0416	0 0296
C_2H_4	0.045	0 1093	0.400	0.330	0.0437	0 0361
CO_2	0.020	0.0765	0.200	0.155	0.0153	0.0119
O	0 0025	0.0070	0.217	0.153	0.0015	0 0119
N	0.0275	0.0670	0.245	0.174	0.0164	0 0117
				Total	0.6955	0 5231

$$n = \frac{0\ 6955}{0.5231} = 1.332$$

TABLE 23
ADIABATIC VALUES OF $\underline{n}$ FOR GAS GIVEN IN TABLE 22, AFTER COMBUSTION

Volume Ratio $\frac{air}{gas}$		6	8	10	12
Combustion of burned gases, Volumes	CO_2	0.530	0.530	0.530	0.530
	H_2O	1.275	1.275	1.275	1.275
	O	0 150	0.570	1.000	1.410
	N	4.768	6.348	7.928	9.508
Constants for burned gases from 1 Lb.	C_p	0.2787	0.2674	0.2623	0.2566
	C_v	0.2035	0.1940	0 1894	0.1851
	$\underline{n}$	1.370	1.380	1.385	1.387

Table 23 gives the data and values after combustion of the different mixtures of Table 22.

The value of n for any particular gas or gaseous mixture is found as in Table 21. In general, rich gases, as illuminating gas or natural gas, require very large ratios of air/gas and therefore have values of n very near the value for pure air, 1.410. In the case of lean gases, as blast furnace gas, or producer gas, requiring small ratios of air/gas, the value of n is lower or nearer the value of n for the gas alone. The limits for ordinary gases and mixtures are between the values of 1. 320 and 1.405, while after combustion these values are lowered somewhat, in general about 1 to 2%.

The range of values for gaseous mixtures is not nearly so large as with steam.

3. VALUES FOR AIR COMPRESSORS.

The proportion of the constituents of air is almost always constant, so that the values C_p and C_v are always constant if we are discussing them at temperatures in the range of compressor practice. This makes the ratio $n = C_p/C_v$ = a constant.

The value of n for air has been determined many times by different investigators in different ways. La Place, in his formula for the velocity of air, which contains n, determined the value to be 1.403. Dulong in experimenting on the oscillations of air in pipes found the value to be 1.421. Wüllner, in some very careful experiments, taking into account the temperature found n to be 1.4053 at 32°, and 1.4029 at 212°.

The degree of variation in determining this constant will be seen from Table 24.

Zeuner after discussing all the values and the experiments supporting them, concludes that the value 1.410 most closely fits the

TABLE 22
ADIABATIC VALUES OF n FOR ILLUMINATING GAS OF TABLE 21 WITH VARIOUS RATIOS $\frac{AIR}{GAS}$

Ratio $\frac{air}{gas}$ by vol.	V	6	8	10	12
Ratio $\frac{air}{gas}$ by Wt	G	15	20	25	30
Wt. per cu ft mixture	Lb.	0.0732	0.0748	0.0757	0.0769
	C_p	0.2667	0.2595	0.2556	0.2527
	C_v	0.1914	0.1858	0.1828	0.1803
	$\underline{n}$	1.393	1.397	1.399	1.402

TABLE 24
ADIABATIC VALUES OF $\underline{n}$ OBTAINED BY VARIOUS EXPERIMENTERS

Experimenter	Value
La Place	1.403
Dulong	1.421
Wüllner @ 32°	1.4053
" @ 212°	1.4029
Clement	1.356
Masson	1.419
Hirn	1.384
Weisbach	1.402
Cazin	1.410
Röntgen	1.405
Lummer and Pringsheim	1.4015

case of pure dry air.

Zeuner also shows that in cases of water injection in compressors, used to keep down the temperature, that the value of n does not change materially from 1.410.

The value, 1.40, is now considered to be the best value obtainable.

4. VALUES OF n FOR COMPRESSED AIR ENGINES.

The temperatures existing in the engines using compressed air for the active medium are not far below those existing in compressors. Within the range covered by both, the value of n is, as far as can be ascertained, practically constant, and will be taken as 1.40.

5. VALUES FOR AMMONIA.

The properties of ammonia have not been as thoroughly investigated experimentally as those of air and the vapor of water. The value of n depending on the value of the ratio C_p/C_v, therefore is not as certain as the values for the other working mediums given. Using the values of $C_p = 0.52$, and $C_v = 0.40$, we get

$$n = 0.52/0.40 = 1.30.$$

This value will be accepted as constant and as the best value obtainable at present. It will be seen that this value, 1.30, is very near to the value adopted for superheated steam, 1.305. The nature of superheated ammonia and steam is very similar.

6. VALUES FOR GAS AND CO_2.

Illuminating gas is often compressed to send it to outlying districts in cities, and then reduce its pressure for distribution. The value of n here depends in each case on the value of the ratio C_p/C_v from the analysis of the particular gas compressed.

The analysis and values of the gas, made by the N. O. Gas Co., New Orleans, La., which is compressed to send it to outlying districts of New Orleans, is given in Table 25.

The writer assisted on a test of Compressor #3 of this plant on Jan. 9, 1909, references to which are given in other parts of this thesis. The value of n for this gas, 1.349, is considerably higher than the value of the gas given in Table 21, 1.332. This is due to the very much larger proportion of CO in the gas of Table 25, and illustrates well the variation of this value with gases of similar heating value, but with a different proportion of the same constituents.

Lummer and Pringsheim give the value for CO_2 as 1.2961 from their experiments. Güldner gives the value as 1.293. This corresponds closely with the experimental value, 1.2961, and will be assumed to be as accurate a value as can be obtained at present.

CHAPTER V.

LABORATORY TESTS.

1. VALUES OF n FOR STEAM OBTAINED UNDER DIFFERENT CONDITIONS WITH THE STEAM ENGINE.

It is plain, from the great range of values of n for steam shown in Chap. III, that the conditions under which these values were obtained must have been subject to wide variation. The engines differed in type, size, speed, steam pressure, steam distribution as affected by valve action, and extent of leakage. Obviously comparisons could not be made of these examples. It was decided, therefore, to study the value of n under known conditions containing as few variables as possible, with the hope that some explanation might be found when the effect of only one variable at a time was studied.

With these requirements and the means at hand, a single cylinder, long-range cut-off, 12 in. X 24 in. Corliss engine, located in the Mechanical Engineering Laboratory of the University of Illinois, was selected for the tests. A description of this engine, the means of loading, the controlling of conditions, and the test methods will be found in Appendix I.

Logarithmic diagrams plotted by the writer, from the same cylinder with different cut-off positions, showed that the value of n was higher as the cut-off was lengthened. There was a large variation in n where the conditions of cylinder size, speed, steam pressure, and steam distribution were the same. The only variable shown by the diagram was the length of cut-off. It was also observed that n was much higher with superheated steam than with saturated steam, under the same general conditions. The only important variable in these two cases to account for this was the quality of the steam mixture at cut-

off, or the proportion of the total weight of mixture in the cylinder which was present as steam at cut-off. All cases examined of superheated steam, at normal cut-off, showed n to be over 1.0, and as high as 1.341, and all cases of small engines using saturated steam showed n to be lower than 1.0, and sometimes as low as 0.70. This led to the conclusion that the quality at cut-off was the largest single controlling factor in the accompanying value of n. Therefore the effort was made to vary the quality at cut-off between the widest practicable limits, and to measure this quality accurately in the tests.

It was desired to study the effect upon the value of n of varying the quality at cut-off, under different conditions of pressure and speed. All tests were run with the steam exhausting from the cylinder at about atmospheric pressure into a surface condenser.

Seventy-four tests were run in 16 series. Of this number 60 tests in 14 series were selected as fulfilling the requirements decided upon to give reliable data. Each series consisted of 4 to 5 separate tests, differing from one another only in the length of cut-off, with the same conditions of pressure and speed. The cut-off was varied in nearly uniform steps from about 5% to 45% of the length of the stroke, and was the means of varying the quality at cut-off, when using either saturated or superheated steam.

The 14 series were divided into 2 divisions of 7 series each, one division being run with saturated steam, and the other with steam superheated to 500°F at the superheater.

Each division consisted of 5 series, run at different gage pressures with constant speed, and 2 series run at different speeds with constant pressure. The steam pressures used were 57.5, 76.5, 95, 113, and 132 lb. gage with the engine running at 120 R.P.M., this being the no-load speed. The other speeds employed were 90 R.P.M. and

TABLE 26 - GENERAL LOG TESTS WITH SATURATED STEAM

				Pressure Lb per sq inch														
Date of Test 1	Series 2	Number of Test 3	Laboratory Designation 4	At Throttle by gage 5	Barometer 6	Absolute Pressure at Throttle 7	Average Absolute Pressure of Series 8	Revolutions per Hour 9	Weight of Condensate per Hour, Lb 10	Weight of Condensate per Revolution, Lb 11	Lb of Steam per Revolution shown by Indicator at Cut-off 12	Lb of Compress'n saved per revolution by Indicator 13	Lb of Steam Mixture Actually present per revolution. 14	Quality of Steam Mixture at Cut-off, 15	Absolute Pressure at Cut-off, Lb 16	Average Value of n from expansion curves 17	Cut off, per cent of Stroke 18	Average Absolute Pressure at Cut off for Series, Lb 19
4-11-10	I	2	5-60-12-120	57.8	14.3	72.1		6428	1054	0.1640	0.0929	0.0248	0.1888	49.2	[illegible]	[illegible]	4.6	
4-16-10		3	16	57.5	14.2	71.5	71.3	6100	11045	0.1811	0.1037	0.0221	0.2032	51.0	61.3	0.882	16.5	60.6
5-12-10		4	32	55.8	14.5	70.3		6302	2010	0.3189	0.2099	0.0276	0.3465	60.6	60.2	0.901	41.2	
4-13-10	II	6	5-80-13-120	77.0	14.3	91.3		6846	1010	0.1475	0.0846	0.0235	0.1710	49.5	70.8	0.927	8.1	
4-16-10		7	25	76.7	14.2	90.9	90.5	6440	1384	0.2149	0.1281	0.0230	0.2379	53.9	80.2	0.895	15.7	77.9
5-12-10		8	38	75.6	14.5	90.1		6640	2104	0.3168	0.2089	0.0253	0.3421	61.0	77.4	0.949	30.9	
5-12-10		9	45	75.0	14.5	89.5		6282	2559	0.4073	0.2727	0.0260	0.4333	62.9	77.4	0.950	42.6	
4-8-10	IV	15	5-100-15-120	95.8	14.3	110.1		6940	1048	0.1510	0.0903	0.0236	0.1746	51.7	95.6	0.910	6.1	
4-16-10		16	34	96.0	14.2	110.2		6510	1530	0.2350	0.1462	0.0226	0.2756	53.1	95.6	0.887	14.6	
4-16-10		17	33	94.9	14.2	109.1	109.1	6432	1584	0.2462	0.1533	0.0233	0.2695	56.9	96.2	0.914	15.5	95.3
5-11-10		18	47	94.5	14.4	108.9		6750	2298	0.3404	0.2145	0.0246	0.3650	58.8	92.5	0.996	25.4	
5-11-10		19	57	93.6	14.4	108.0		6410	2758	0.4302	0.2952	0.0263	0.4565	64.7	95.6	1.002	37.0	
5-10-10		20	60	94.0	14.4	108.4		5958	3008	0.5048	0.3484	0.0248	0.5296	65.8	96.1	0.963	44.3	
4-11-10	V	22	5-120-16-120	114.6	14.3	128.9		7046	1030	0.1461	0.0907	0.0239	0.1700	53.3	105.5	0.835	4.8	
4-16-10		23	34	113.7	14.2	127.9	127.7	6688	1566	0.2341	0.1445	0.0240	0.2581	56.0	110.0	0.944	11.3	109.4
4-16-10		24	40	113.6	14.2	127.8		6466	1738	0.2687	0.1776	0.0227	0.2914	61.0	109.1	0.940	15.7	
4-29-10		25	62	112.0	14.2	126.2		6736	2664	0.3954	0.2708	0.0253	0.4207	64.4	113.0	1.004	27.3	
4-11-10	VI	27	5-140-16-120	131.9	14.3	146.2		7086	986	0.1391	0.0941	0.0240	0.1631	57.6	126.7	0.946	3.7	
4-16-10		28	34	133.2	14.2	147.4		6862	1466	0.2136	0.1426	0.0231	0.2367	60.2	130.8	0.950	8.3	
4-16-10		29	49	132.0	14.2	146.2	146.2	6306	2000	0.3172	0.2050	0.0220	0.3392	60.4	130.0	1.006	16.4	129.4
4-29-10		30	68	130.8	14.2	145.0		6840	2796	0.4087	0.2760	0.0256	0.4343	63.5	128.9	1.008	24.9	
5-10-10		31	77	132.0	14.4	146.4		6138	3276	0.5337	0.3762	0.0269	0.5606	67.1	130.6	1.010	35.3	
5-9-10	VII	32	5-120-14-90	113.9	14.4	128.3		5258	1025	0.1949	0.0909	0.0251	0.2200	41.3	112.1	0.854	4.9	
5-9-10		33	30	113.9	14.4	128.3	128.4	5102	1585	0.3106	0.1641	0.0242	0.3348	49.0	113.2	0.932	12.8	113.7
5-2-10		34	41	114.1	14.4	128.5		4940	2090	0.4230	0.2402	0.0246	0.4476	53.7	115.7	0.930	23.2	
5-2-10	VIII	36	5-120-28-150	113.3	14.4	127.7		8772	1616	0.1842	0.1136	0.0260	0.2102	54.0	108.4	0.927	7.7	
"		37	55	114.2	14.4	128.6	127.9	8518	2554	0.2998	0.1990	0.0254	0.3252	61.2	110.3	0.964	19.7	110.3
"		38	78	112.8	14.4	127.2		8182.5	3520	0.4302	0.3080	0.0256	0.4558	67.6	110.1	1.045	34.4	
"		39	91	113.5	14.4	127.9		7625	4142.5	0.5433	0.3968	0.0259	0.5692	69.7	112.2	1.016	46.0	

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TABLE 27 - GENERAL LOG. TESTS WITH SUPERHEATED STEAM

Date of Test 1	Series 2	Number of Test 3	Laboratory Designation 4	Pressure Lb per sq inch				Revolutions per Hour 9	Weight of Condensate per Hour, Lb 10	Weight of Condensate per Revolution, Lb. 11	Lb of Steam per Revolution shown by Indicator at Cut-Off 12	Lb of Compression saved per revolution by Indicator 13	Lb of Steam Mixture Actually present per revolution 14	Quality of Steam Mixture Cut-off 15	Absolute Pressure of Cut-off, Lb. 16	Average Value of n from expansion curves 17	Cut-off, per cent of Stroke 18	Average Absolute Pressure at Cut-off for Series, Lb. 19
				At Throttle by gage 5	Barometer 6	Absolute Pressure at Throttle 7	Average Absolute Pressure of Series 8											
2-17-11	IX	40	500°-60-8-120	59.5	14.5	74.0		7040	815	0.1158	0.0838	0.0232	0.1390	60.2	61.2	0.904	11.7	
"		41	16	60.5	14.5	75.0	73.4	6856	1058	0.1543	0.1235	0.0239	0.1782	69.3	61.6	1.053	21.6	61.1
"		42	24	57.7	14.5	72.2		6522	1288	0.1975	0.1738	0.0228	0.2203	78.8	60.6	1.114	32.0	
"		43	28	57.8	14.5	72.3		6262	1396	0.2229	0.2110	0.0227	0.2456	85.9	60.8	1.144	41.3	
2-18-11	X	44	500°-80-10-120	76.1	14.5	90.6		6655	1201.7	0.1806	0.1106	0.0246	0.2052	53.9	79.2	0.951	12.4	
"		45	21	77.4	14.5	91.9	91.0	6550	1294	0.1976	0.1342	0.0250	0.2226	60.3	79.2	0.957	16.5	78.5
"		46	31	76.3	14.5	90.8		6290	1455	0.2313	0.1910	0.0230	0.2543	75.1	77.4	1.105	27.6	
"		47	40	76.0	14.5	90.5		5908	1822	0.3084	0.2720	0.0226	0.3310	82.2	78.0	1.112	41.5	
2-17-11	XI	48	500°-100-13-120	94.3	14.5	108.8		7112	822	0.1156	0.0877	0.0228	0.1384	63.4	92.8	0.924	6.1	
"		49	26	93.0	14.5	107.5		6936	1270	0.1831	0.1311	0.0234	0.2065	63.5	90.7	0.982	13.3	
"		50	39	95.5	14.5	110.0	109.0	6682	1660	0.2484	0.2040	0.0226	0.2710	75.3	97.1	1.040	22.3	94.1
2-20-11		51	45	94.7	14.4	109.1		6500	1780	0.2738	0.2479	0.0220	0.2958	83.8	96.4	1.116	28.8	
"		52	52	95.1	14.4	109.5		6240	2072	0.3321	0.2821	0.0226	0.3547	79.6	93.6	1.001	39.2	
2-20-11	XII	53	500°-120-16-120	115.2	14.4	129.6		7100	914	0.1287	0.0972	0.0224	0.1511	64.2	111.5	0.991	5.1	
"		54	32	114.3	14.4	128.7		6932	1380	0.1991	0.1561	0.0225	0.2216	70.5	109.0	0.990	13.0	
"		55	48	114.1	14.4	128.5	128.4	6662	1906	0.2861	0.2419	0.0226	0.3087	78.3	111.7	1.032	22.9	111.0
1-7-11		56	60	114.1	14.3	128.4		6326	2315	0.3660	0.3230	0.0219	0.3879	83.3	111.5	1.140	35.1	
2-18-11		57	62	112.3	14.5	126.8		5940	2350	0.3956	0.3709	0.0228	0.4184	88.6	111.5	1.209	40.3	
1-7-11	XIV	61	500°-140-17-120	132.5	14.3	146.8		7136	922	0.1292	0.0935	0.0228	0.1520	61.5	127.7	0.939	3.2	
1-7-11		62	34	128.0	14.3	142.3		7010	1384	0.1974	0.1553	0.0229	0.2203	71.5	123.6	0.969	11.0	
"		63	51	131.3	14.3	145.6	145.4	6810	1789	0.2627	0.2331	0.0228	0.2855	81.7	131.0	1.074	18.1	128.5
1-10-11		64	63	131.0	14.3	145.3		6602	2171	0.3288	0.3113	0.0222	0.3510	88.7	130.5	1.190	26.9	
2-18-11		65	73	132.6	14.5	147.1		6088	2640	0.4326	0.3988	0.0230	0.4566	87.3	129.5	1.165	37.1	
2-17-11	XV	66	500°-120-12-90	114.8	14.5	129.3		5322	808	0.1518	0.0950	0.0220	0.1738	54.6	112.3	0.948	4.6	
"		67	24	114.2	14.5	128.7	128.0	5212	1218	0.2337	0.1540	0.0222	0.2559	60.2	111.1	0.941	12.4	111.7
"		68	36	112.3	14.5	126.8		5018	1574	0.3137	0.2356	0.0226	0.3363	70.0	110.7	1.013	23.1	
"		69	43	112.7	14.5	127.2		4817	1809	0.3755	0.3078	0.0217	0.3972	77.4	112.5	1.148	32.0	
2-17-11	XVI	71	500°-120-36-150	112.8	14.5	127.3		8700	1512	0.1738	0.1447	0.0249	0.1987	72.9	108.0	1.093	12.3	
"		72	54	111.8	14.5	126.3	126.9	8438	2018	0.2392	0.2080	0.0257	0.2649	78.6	109.0	1.091	20.5	108.3
"		73	70	112.2	14.5	126.7		8084	2458	0.3041	0.2970	0.0236	0.3277	90.7	107.2	1.213	32.4	
		74	82	112.6	14.5	127.1		7494	3110	0.4150	0.4135	0.0237	0.4387	94.3	108.8	1.234	46.9	

TABLE 28 LOG OF GROUP AVERAGES

Group	Series	Number of Tests Inclusive 3	Laboratory Designation 4	Pressure Lb per sq inch: Average Absolute Pressure at Throttle by Gage 5	Average Absolute Pressure of Group 6	Difference for Group Average of next lower Pressure 7	Absolute pressure from Indicator diagram Lb per sq inch: Average at Cut-off 8	Average of Group 9	Difference from Group Average of next lower Pressure 10	Drop in Pressure Throttle to Cut-Off 11	Pressure Adopted to find Quality of Steam Mixture at Cut-off 12	Specific Volume of Steam at Pressure Adopted 13	Number of Tests in Series 14	Number of Tests in Group 15	Number of Diagrams Analyzed 16	Average Quality of Steam Mixture at Cut-off from Tests in Group 17	Average Value of n from Tests in Group 18
A	I	2-4	5-60- -120	71.3	72.4		60.6	60.9		11.5	61.0	[illegible]	3	7	14	65.00	0.998
	IX	40-43	500°-60- -120	73.4			61.1						4				
B	II	6-9	5-80- -120	90.5	90.8	18.4	77.9	78.2	17.3	12.6	78.0	5.60	4	8	16	62.35	0.981
	X	44-47	500°-80- -120	91.0			78.5						4				
C	III	15-20	5-100- -120	109.1	109.1	18.3	95.3	94.7	16.5	14.4	95.0	4.65	6	11	22	65.15	0.976
	XI	48-52	500°-100- -120	109.0			94.1						5				
D	V	22-25	5-120- -120	127.7	127.9	18.8	109.4	110.7	16.0	17.2	111.0	4.012	4	9	18	68.85	1.010
E	VI	32-34	5-120- -90	128.4			113.7						3	7	14	58.03	0.967
F	VII	36-39	5-120- -150	127.9			110.3						4	8	16	73.65	1.073
D	XII	53-57	500°-120- -120	128.4			111.0						5				
E	XIV	66-69	500°-120- -90	128.0			111.7						4				
F	XV	71-74	500°-120- -150	126.9			108.3						4				
G	IV	27-31	5-140- -120	146.2	145.8	17.9	129.4	129.0	18.3	16.6	129.0	3.478	5	10	20	69.75	1.026
	XIII	61-65	500°-140- -120	145.4			128.5						5				

and 150 R.P.M. at the gage pressure of 113 lb. This gave the effect of the use of 5 steam pressures at one speed, and 3 speeds at one pressure.

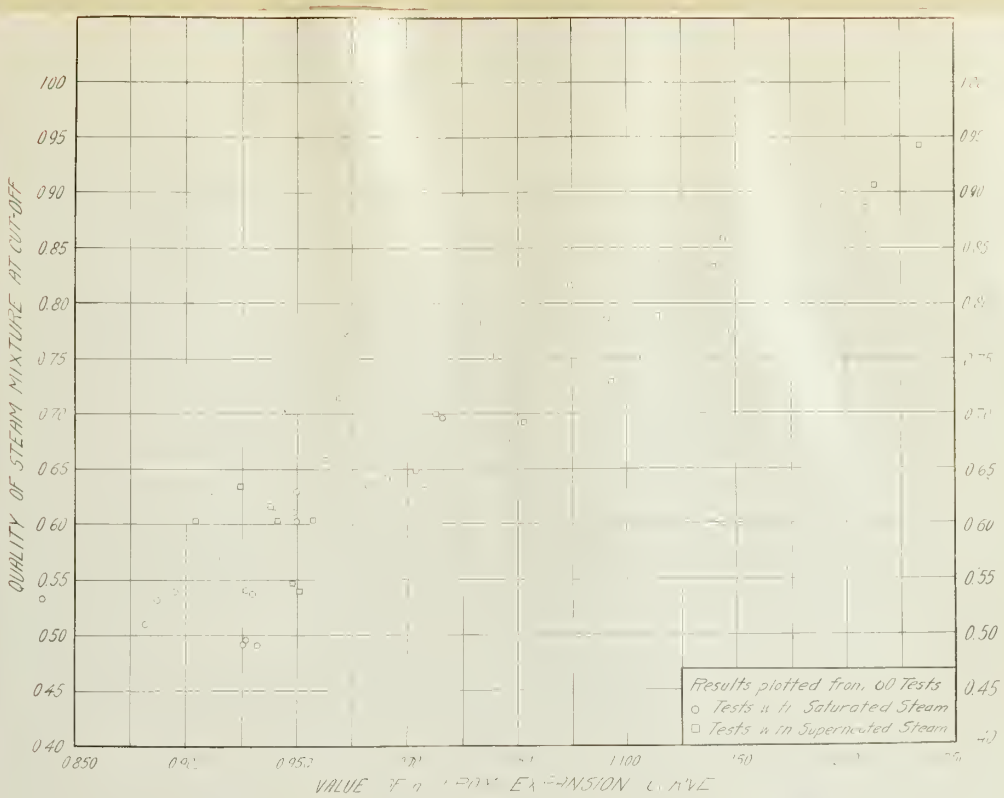
Wide variation of the quality at cut-off was obtained by the use of saturated and superheated steam, in conjunction with different lengths of cut-off, under the same conditions of pressure and speed. The qualities obtained ranged from about 0.50 to 0.90, covering the range usually found in practice with the type of engine tested.

The qualities at cut-off were found by the method shown in Appendix II. The values of n were obtained by the logarithmic method described in Chap. III. The values of both quality and n given are the average of the results obtained from one set of head end and crank end diagrams for each test, selected in the manner described in Appendix II.

The general log of the 29 tests with saturated steam is shown in Table 26. Table 27 contains the results of the 31 tests with superheated steam. Table 28 shows the averages of similar series, called groups, run at the same pressure and speed, with both saturated and superheated steam.

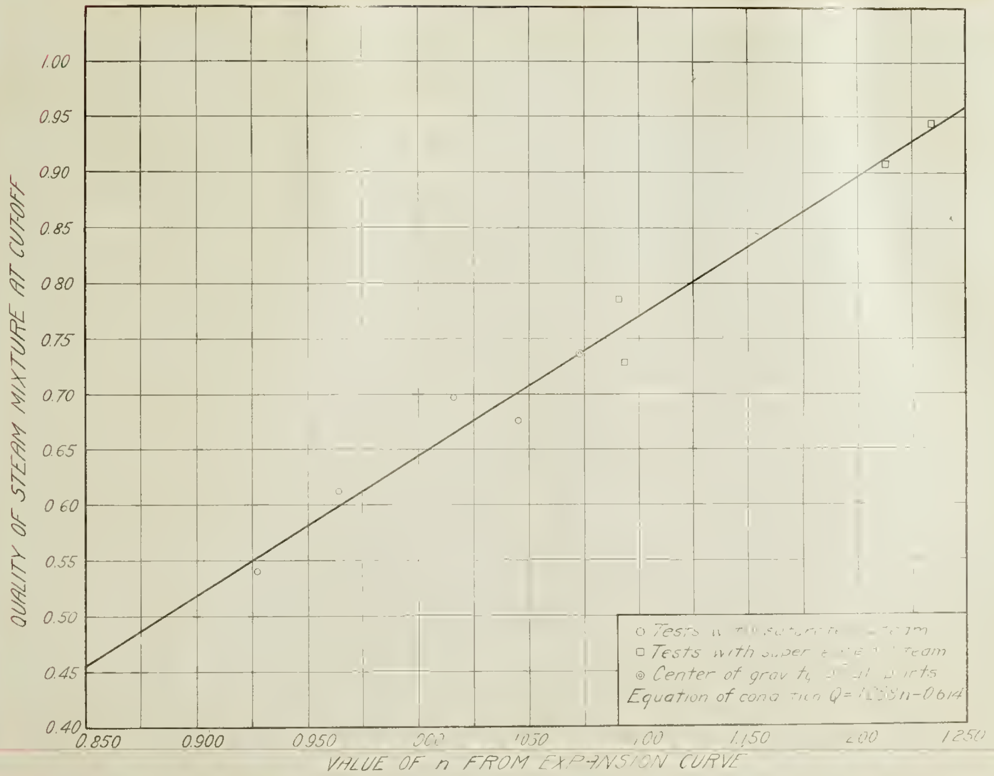
For any one group of two series of tests, as has already been pointed out, the only variables are the length of the cut-off, and the quality at cut-off, which will be called Q in subsequent discussion.

All the simultaneous values of Q and n obtained from the 60 tests are plotted in Fig. 4. A study of this figure shows beyond question that as Q increases in value, n increases also. The values all lie in a region which has a definite trend towards higher simultaneous values of Q and n. Observing the general trend, it is seen that there is no exception to this general relation. No value of n below



GENERAL RELATIONS BETWEEN QUALITY AT CUT-OFF AND [illegible]
FOR VARIOUS PRESSURES AND SPEEDS

Fig. 4.



RELATION OF QUALITY AND THE VALUE OF n FOR TESTS
RUN AT 111 LB ABSOLUTE PRESSURE OF CUTOFF AND 150 R.P.M.

Fig. 5.

1.000, for instance, is found for values of Q above 0.80, and no value of n above 1.100 is found for values of Q below 0.72.

It is also apparent that the points with late cut-off, obtained from saturated steam for given values of Q and n, are in the same region occupied by the points with early cut-off obtained from superheated steam for the same given values of Q and n. Examining the region of n = 0.900 - 1.000 and of Q = 0.55 - 0.70, we see that the points obtained with late cut-off with saturated steam, and with early cut-off with superheated steam, lie indiscriminately together. This shows conclusively, in a general way, that the value of n is practically independent of the cut-off position, even though this may vary from 5% to 45%, and that it depends solely on the value of Q.

The points shown in Fig. 4 occupy a relatively wide region until they are separated into the various groups of similar pressures and speeds.

The points for each group were plotted separately, and separate curves were determined for each condition. From preliminary plotting, the relations were found to be expressed closer by straight lines than any other family of curves. The method used to draw these lines will be given for Group F, which includes Series VIII, with 4 tests, and Series XVI with 4 tests. This is shown in Fig. 5. All points were given equal weight. The average of all the coordinates, or the "center of gravity", was found and the condition imposed that the line pass through this center as an axis, and that the slope be determined by the position of the points. The points in Group F were divided into four logical groupings, and the center of gravity found for each grouping. The line was then drawn as shown. Where points were located so that logical grouping was in doubt, various groupings were made, and each given weight in determining the slope.

The equation of the curve selected is

$$Q = 1.258\,n - 0.614.$$

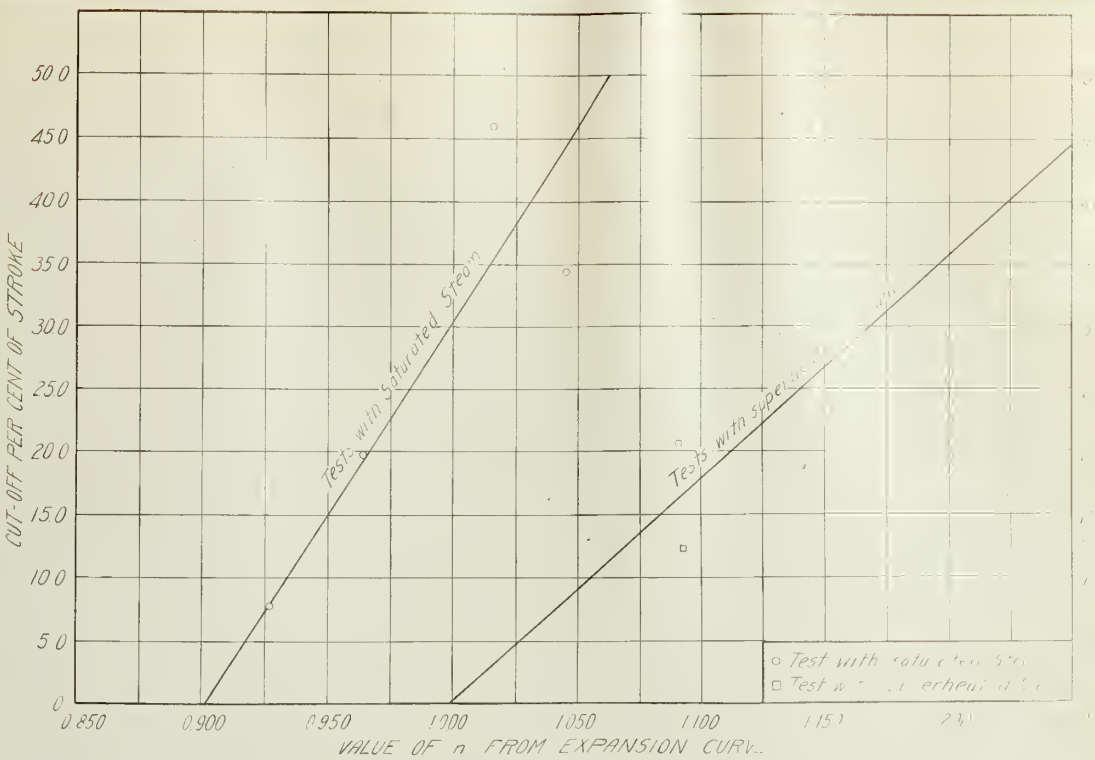
The average deviation of the points from this line is 2.6%(measured from the zero of Q), and the maximum deviation is 4.6%. This average deviation, 2.6%, is smaller than that for most of the groups. The straight line, in most cases, represents the points found as close as any other curve that could be employed, and has the merit of simplifying greatly the subsequent use made of the relations for the different groups.

The values of n for Group F were also plotted to the various accompanying cut-off positions at which each test was run. This is shown in Fig. 6. These points, by their position, show definitely for one group what has been shown in general in Fig. 4, that n is not a function of the cut-off value within the limits examined.

a. Effect of Varying the Steam Pressure at Constant Speed.

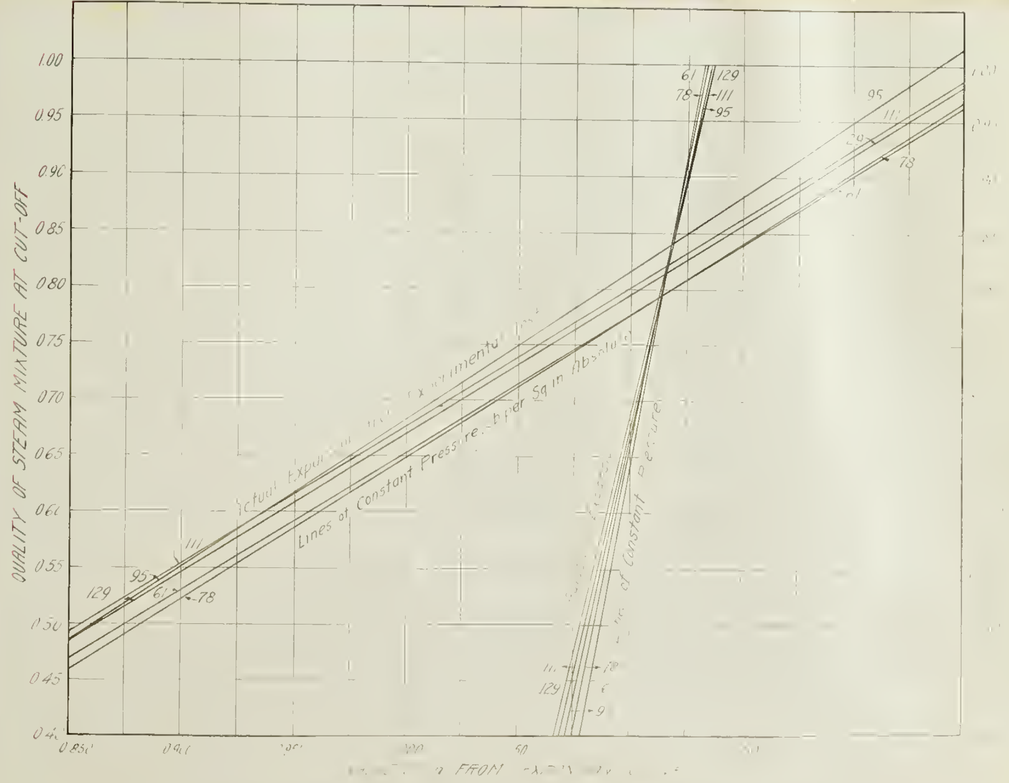
The relations of Q and n were determined separately for all groups by the method outlined for Fig. 5. The lines for the five pressures used, comprising the results of Groups A, B, C, D, and G were replotted as shown in Fig. 7. This figure also contains other curves that will be discussed in Chap. VI. The lines shown give the relations of Q and n for various absolute pressures at cut-off, all obtained with a speed of 120 R.P.M., measured at no-load.

These curves were then examined to find the effect of varying the cut-off pressure(designated as P) on the relations of Q and n. In Fig. 7, the constant pressure curves were intercepted at constant values of n, and the coordinates of Q and P for the points of intersection plotted in Fig. 8. This process was repeated at intervals of 0.05, for the values of n, from 0.850 to 1.250. The resulting points did not give smooth curves, but after considerable study and trial, curves were adopted as shown. The method used was as follows: the



RELATIONS OF PERCENT OF CUT-OFF AND THE VALUE OF n FOR TESTS RUN AT 111 LB. ABSOLUTE PRESSURE AT CUT-OFF AND 150 R.P.M.

Fig. 6.



RELATIONS [illegible] BETWEEN QUALITY [illegible]

Fig. 7.

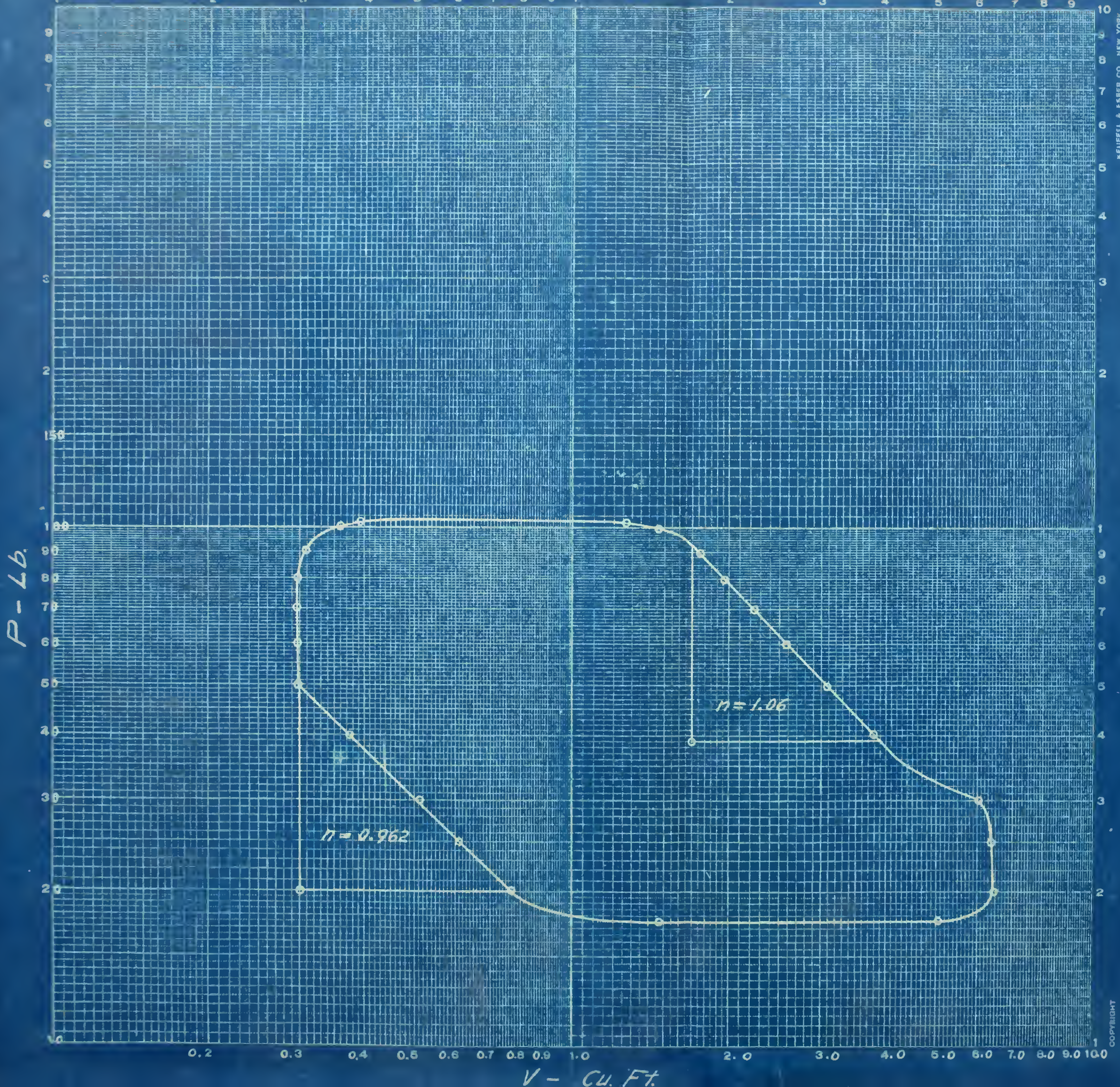


Fig. 11
Greene 18¼" x 43"
Steam Engine

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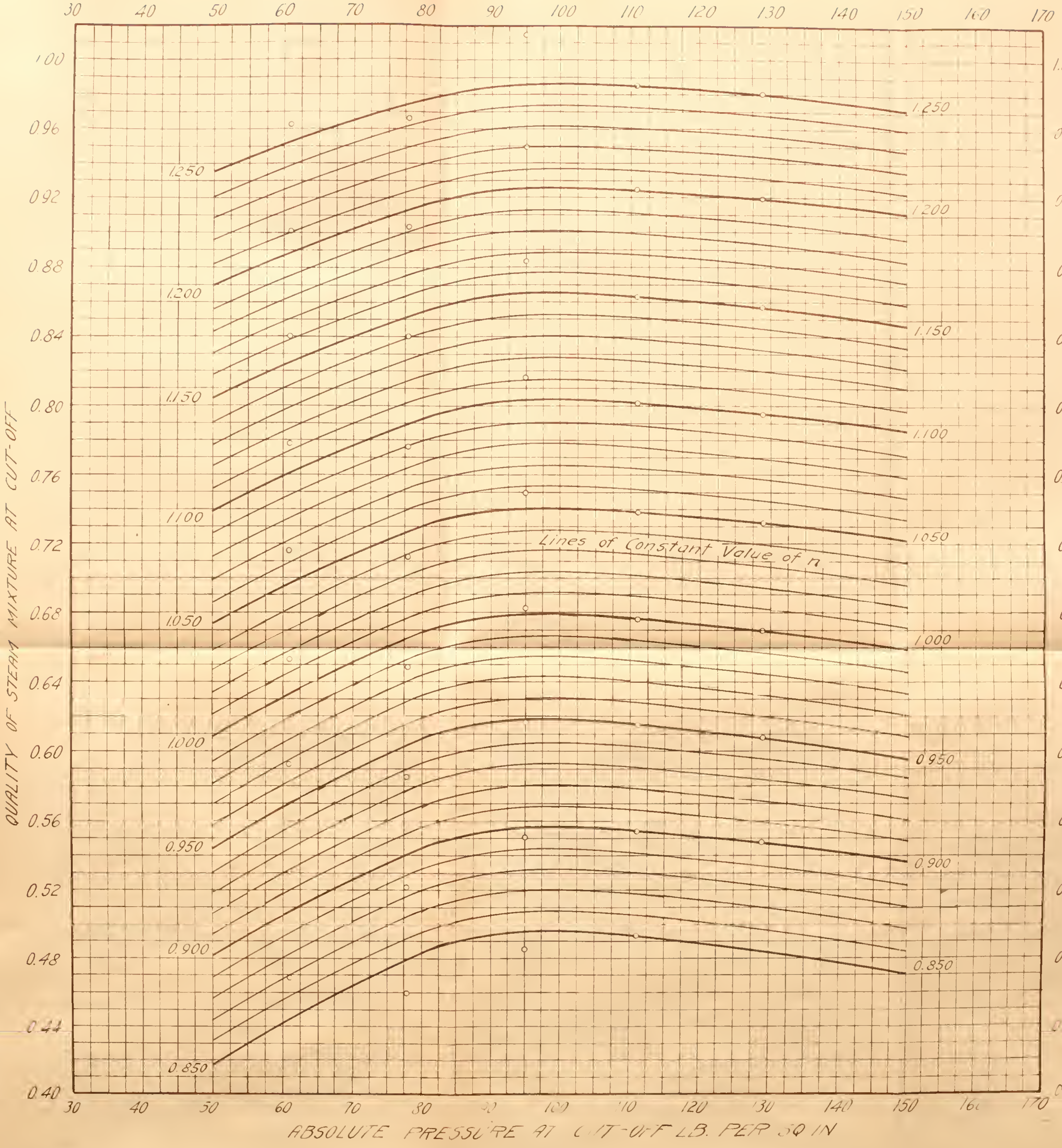


CHART SHOWING THE RELATION AT CUT-OFF, BETWEEN QUALITY AND PRESSURE FOR CONSTANT VALUES OF n FROM EXPANSION CURVE maximum value, from the evidence of Fig. 7, was assumed to be at the value, $P = 95$; the values for $P = 61$ and $P = 78$ were combined, and the center of the line connecting each pair used as one point; with these assumptions the curves were drawn. Since the relation of Q and n at constant speed and pressure is a straight line, the lines on Fig. 8 were drawn by interpolation to increments in the value of n to 0.01.

This procedure gave a series of relations between Q and P for constant values of n. Since, however, the independent variables, in any actual curve under examination, are n and P, the coordinates of the curves of Fig. 8 were changed so as to show the relations of n and P at constant values of Q. These relations are shown in Chap. VII as Fig. 30.

The effect of pressure on the relations of Q and n is not great between the limits of 75 to 150 lb. An approximate equation has been worked out, therefore, which represents the relations of Q and n at an average pressure between the limits mentioned. The equation corresponds to the relations at 129 lb. and is of the form

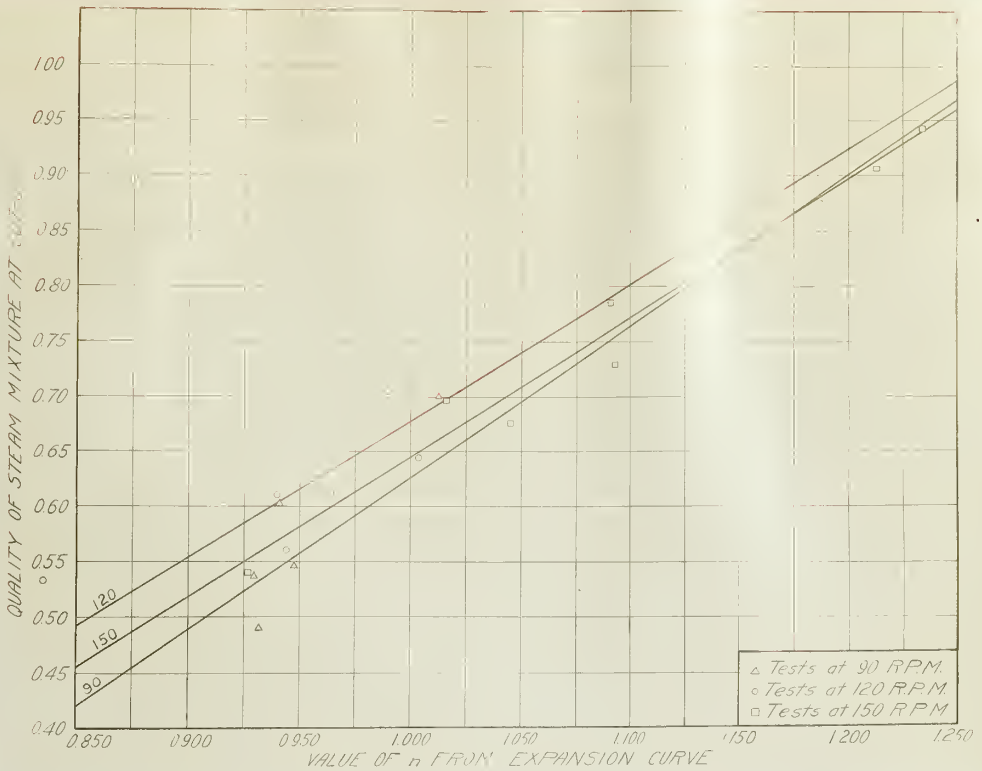
$$Q = 1.245\,n - 0.576$$

b. Effect of Varying the Speed at Constant Pressure.

Groups D, E, and F were run at speeds of 120, 90, and 150 R.P.M. respectively,(all speeds being measured at no-load) with the cut-off pressure on the diagrams constant at 111 lb. absolute.

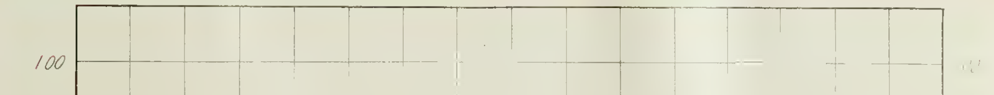
The regulation of the governor was very poor, there being about a 10% drop in speed from no-load to full load. For this reason, the relations of Q and n with various speeds at constant pressure were affected by considerable variation of the speed it self for each group.

The relations for each group were found as already described, and the curves plotted in Fig. 9. The relation of speed(designated as S) and Q for constant values of n was derived from Fig. 9, and is given in Fig. 10.



RELATION OF QUALITY AT CUT-OFF AND THE VALUES OF n FOR VARIOUS SPEED OF THE CONSTANT CUT-OFF PRESSURE [illegible]

Fig. 9.



RELATIONS, AT CUT-OFF, BETWEEN QUALITY AND SPEED FOR CONSTANT VALUES OF n AT CONSTANT PRESSURE

Fig. 10.

The apparent relations of Q, S, and n, obtained by drawing a smooth curve through the three points obtained for each value of n, is not satisfactory owing to insufficient data and the change of the speed itself in the three groups due to poor regulation.

The drop in the speed, for one group, does not seriously affect, however, the relations of Q and n for the various pressures at constant speed.

Reliable values of Q are also relatively difficult to obtain at speeds as low as 90 R.P.M.

2. RELATION OF THE VALUES OF n TO THE QUALITY OF THE STEAM MIXTURE AT CUT-OFF.

From the evidence obtained from these tests, it may be stated positively that, for any one engine, running at a given pressure and speed, there is a definite relation between Q and n which is practically independent of the cut-off position. This relation is apparently a linear one. It may also be stated that the relation of Q and n is dependent, to some extent, on the absolute pressure at cut-off, and the speed of the engine.

After all these tests were made, the writer, in preparing Chap. IV, found, to his surprise, that the value of n for adiabatic expansion varied considerably with the initial quality, and, to a smaller extent, with the initial pressure. When this fact was known, it seemed entirely reasonable that the same law should hold in cylinders, but perhaps greatly modified and exaggerated by the action of the cylinder walls.

It only remained to compare the relations of Q and n for the engine tested with those for other engines. This comparison is given in Chap. VI.

CHAPTER VI.

DISCUSSION OF CYLINDER ANALYSIS BY THE NEW METHOD.

1. DOES THE LAW $PV^n = C$ HOLD FOR CURVES FROM PRACTICE?

It has long been known that adiabatic expansion, or compression, of any elastic medium takes place substantially according to the law $PV^n = C$. The values of n are different for different media, and sometimes vary for the same medium, with different conditions of the initial state. These values, for media commonly used in reciprocating engines, are given in Chap. IV.

In practice, however, expansion is never adiabatic, but is changed in character by the presence of the metal surrounding the working medium, and by imperfection of mechanism. Since the actual change of state is not adiabatic, the question arises as to how it has been changed in character, and whether this modified expansion still obeys the law $PV^n = C$.

Many investigators have examined the curves from actual diagrams to clear up this point for steam. These include Zeuner, Leloutre, Lüders, Perry, Hirn, and Dwelshausers-Dery. They find that expansion in steam cylinders takes place substantially according to the law $PV^n = C$, but that n varies in value between wide limits in different cases.

The writer has examined about 300 diagrams from eninges using steam, gas, air and ammonia to investigate this point. As a result, it may be stated that, in the case of the great majority of engines using elastic media, expansion and compression take place substantially according to the law $PV^n = C$. Certain exceptions, however, have been found, and the causes studied. These causes are treated in the next section of this Chapter.

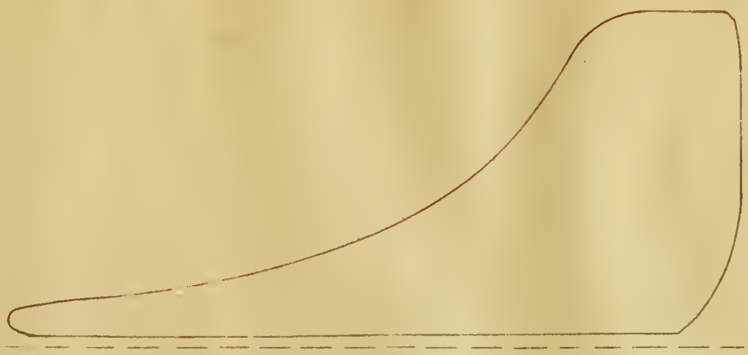


Fig. 11
Crank End Scale 50#

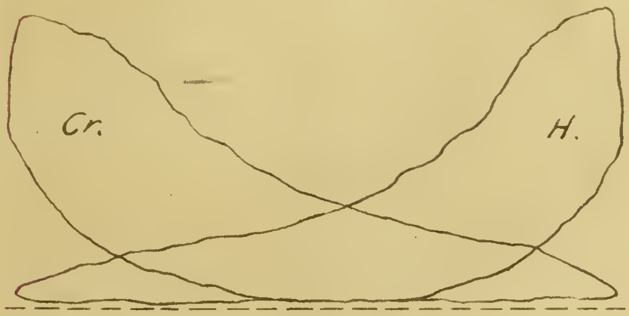


Fig. 12
Scale 80#

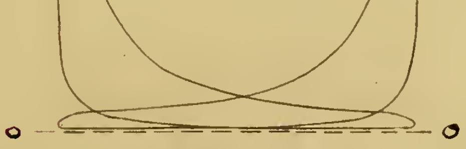


Fig. 15
Scale 162#

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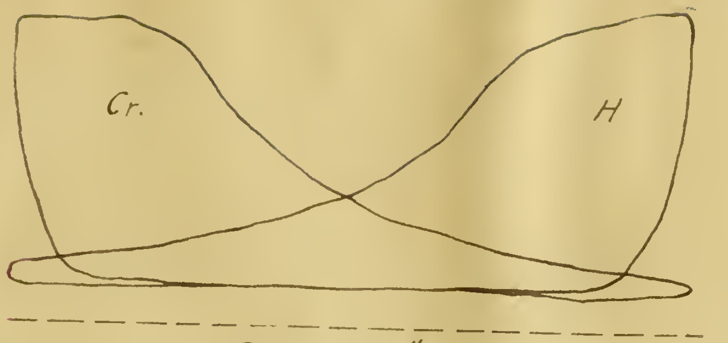
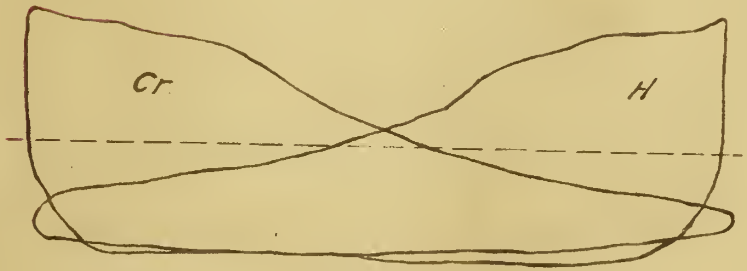


Scale 80#
High Pressure



Scale 20#
Low Pressure
Fig. 13

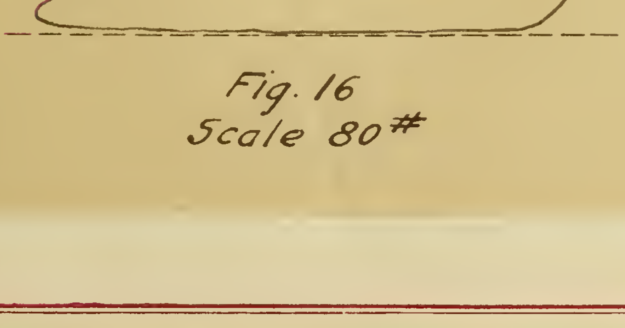
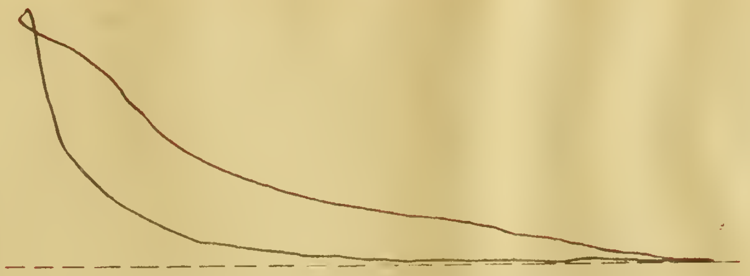
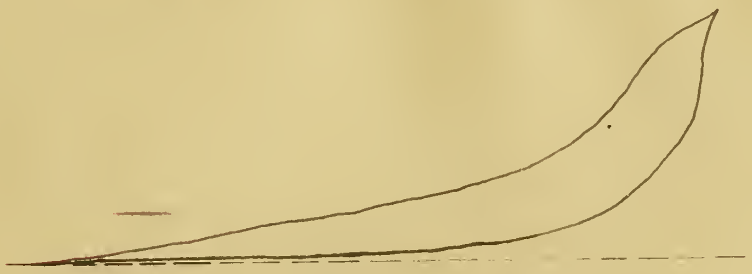


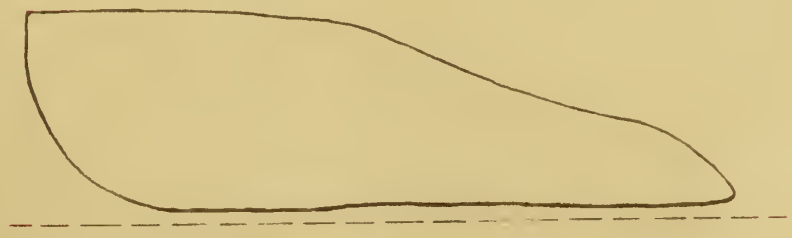
Fig. 16
Scale 80#



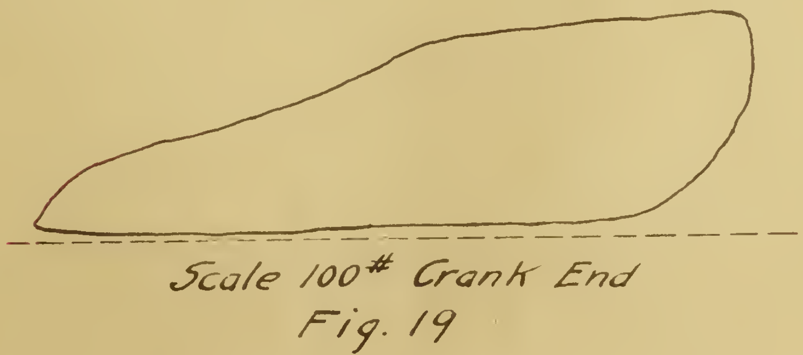
Scale 150# Head End



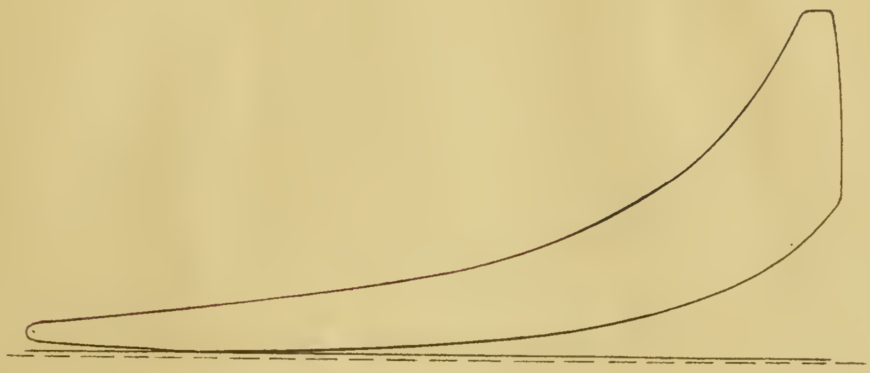
Scale 150# Crank End
Fig 17



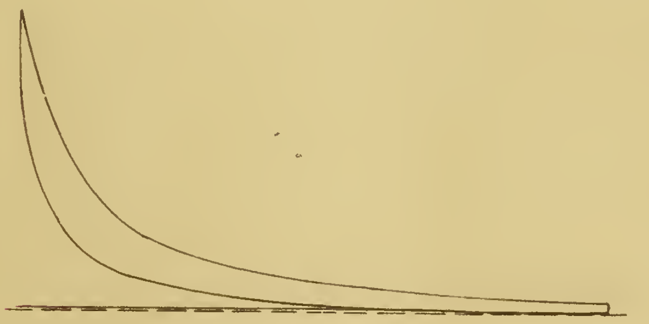
Scale 100# Head End



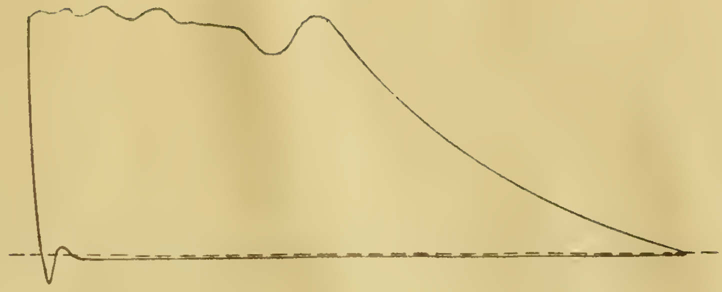
Scale 100# Crank End
Fig. 19



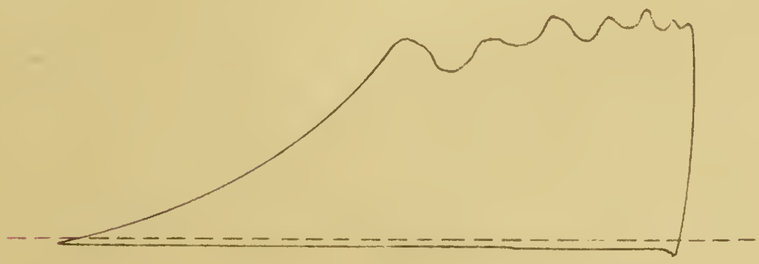
Scale 160#
Fig. 20



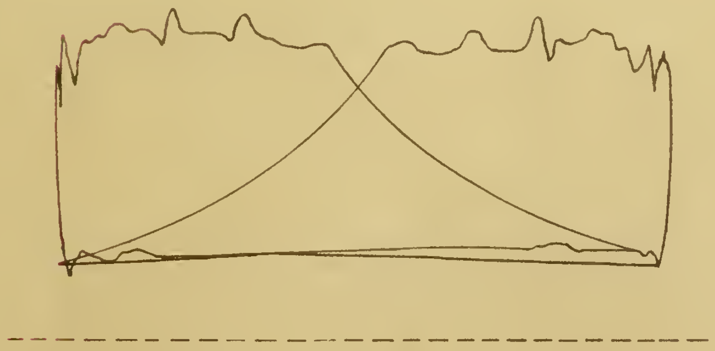
Scale 400#
Fig. 21



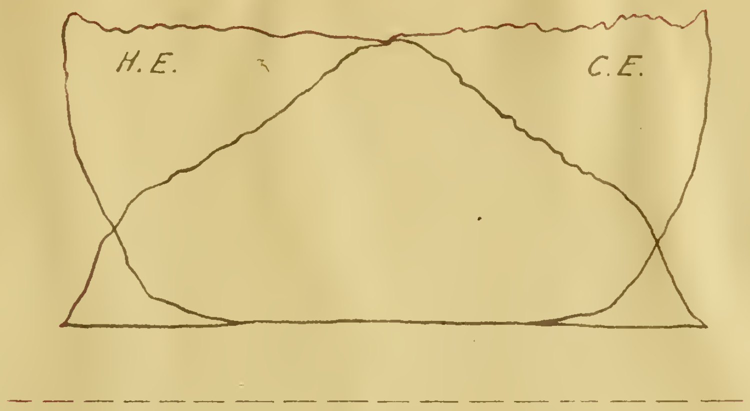
Head End Low Pressure
Scale 17.5#



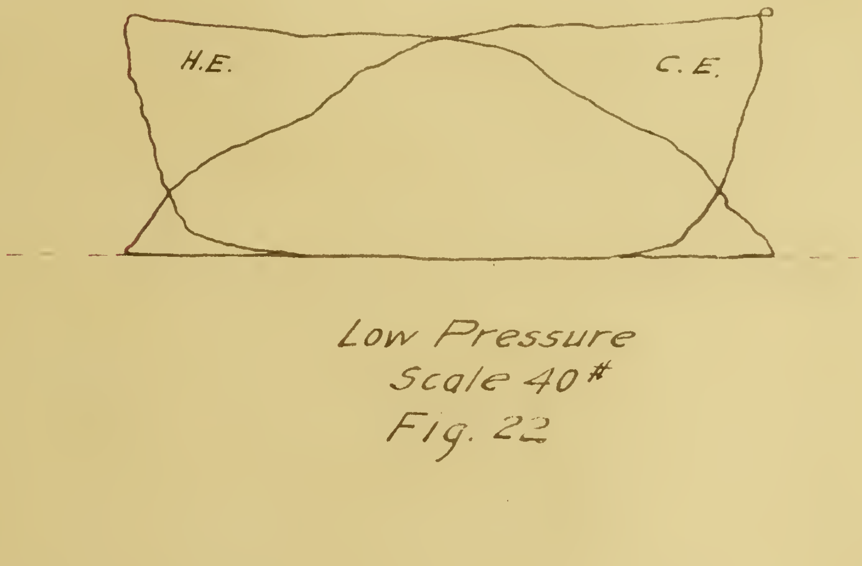
Crank End Low Pressure
Scale 21.6#



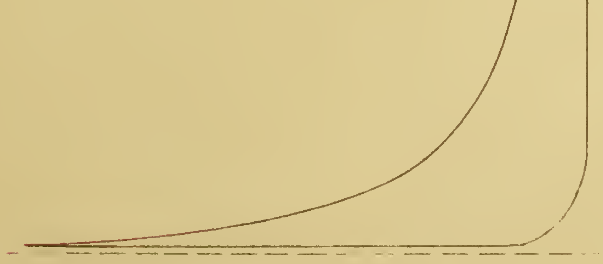
High Pressure
Scale 50#
Fig. 23



High Pressure
Scale 120#



Low Pressure
Scale 40#
Fig. 22



Scale 136#
Fig 24

P- Lb.

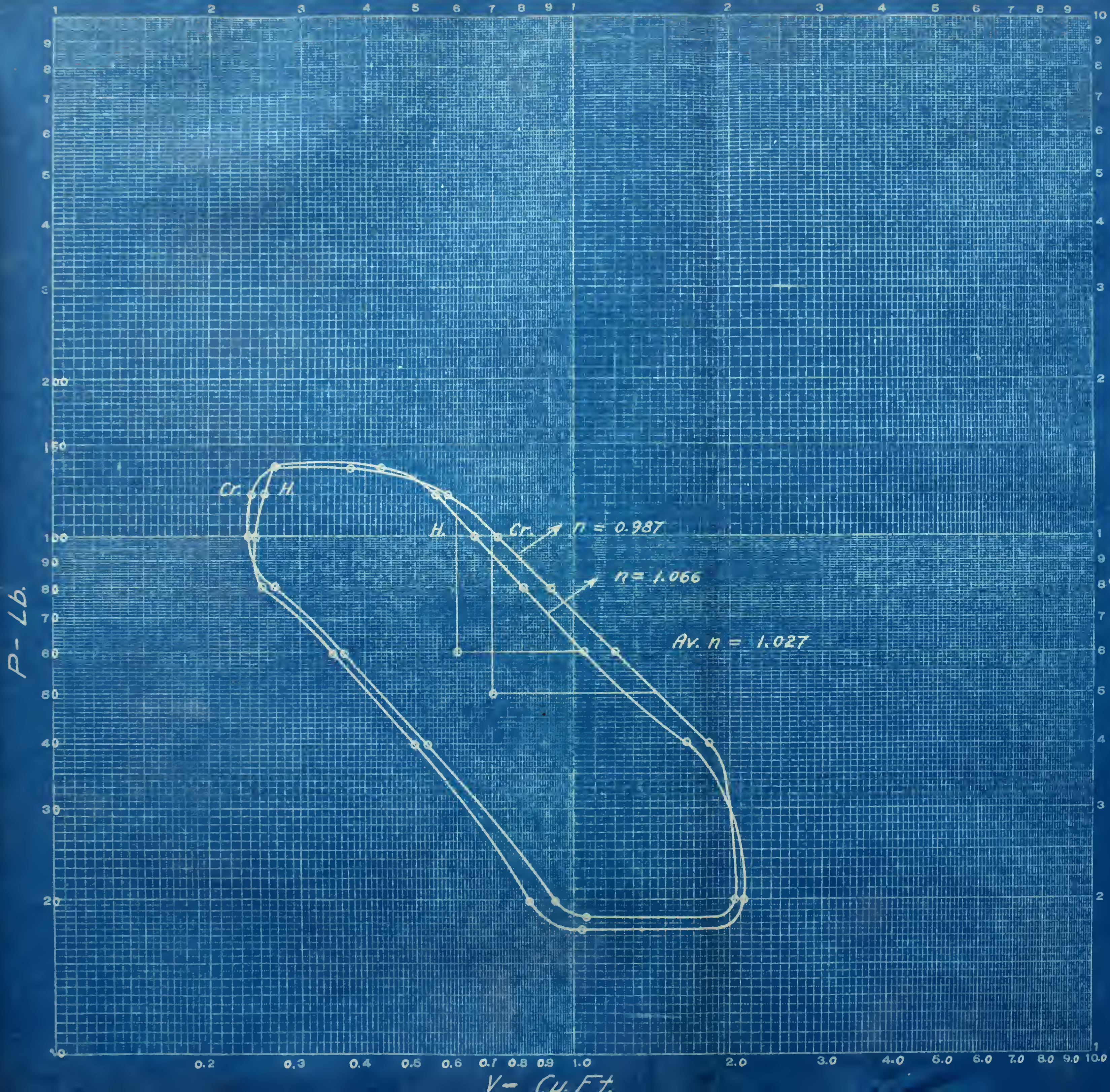


Fig. 12
Ide 16"x16" High
Speed Steam Engine

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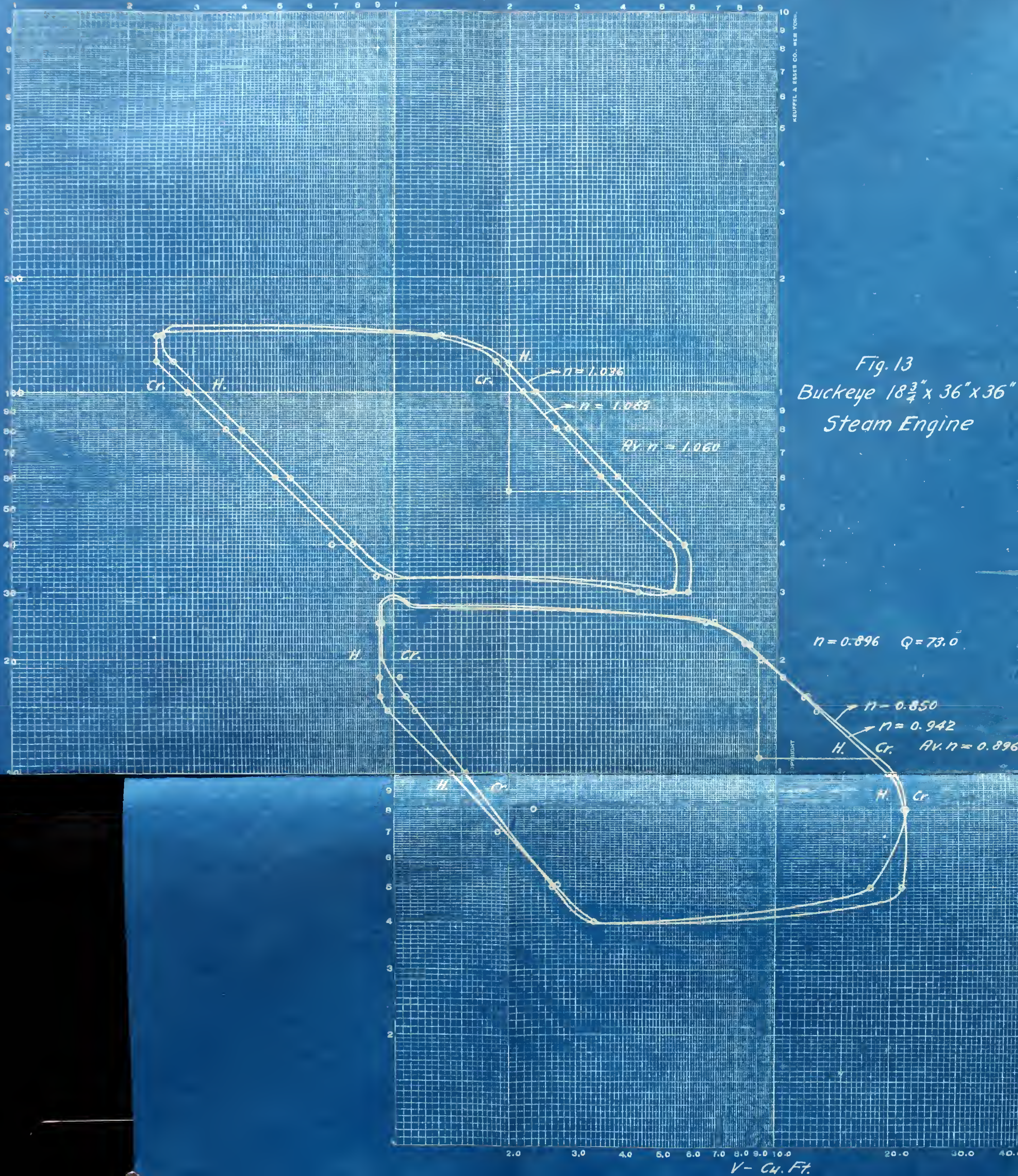

Fig. 13
Buckeye 18¾" x 36" x 36"
Steam Engine

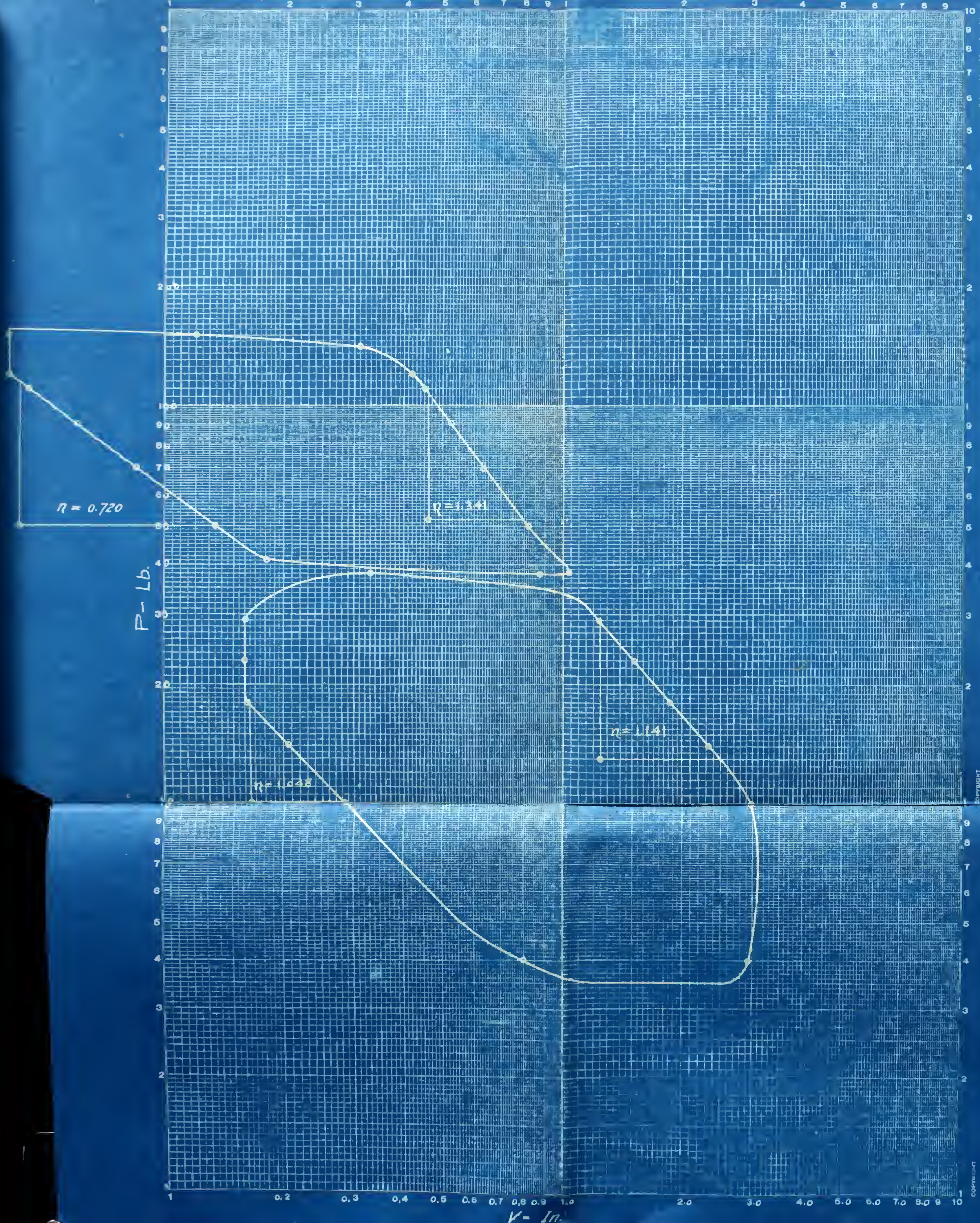

Fig. 14

Rice and Sargent 16" x 28" x 42
Steam Engine using highly
Superheated Steam.
Test No. 1 A.S.M.E. XXV, 267
See this book for indicator
Diagrams.

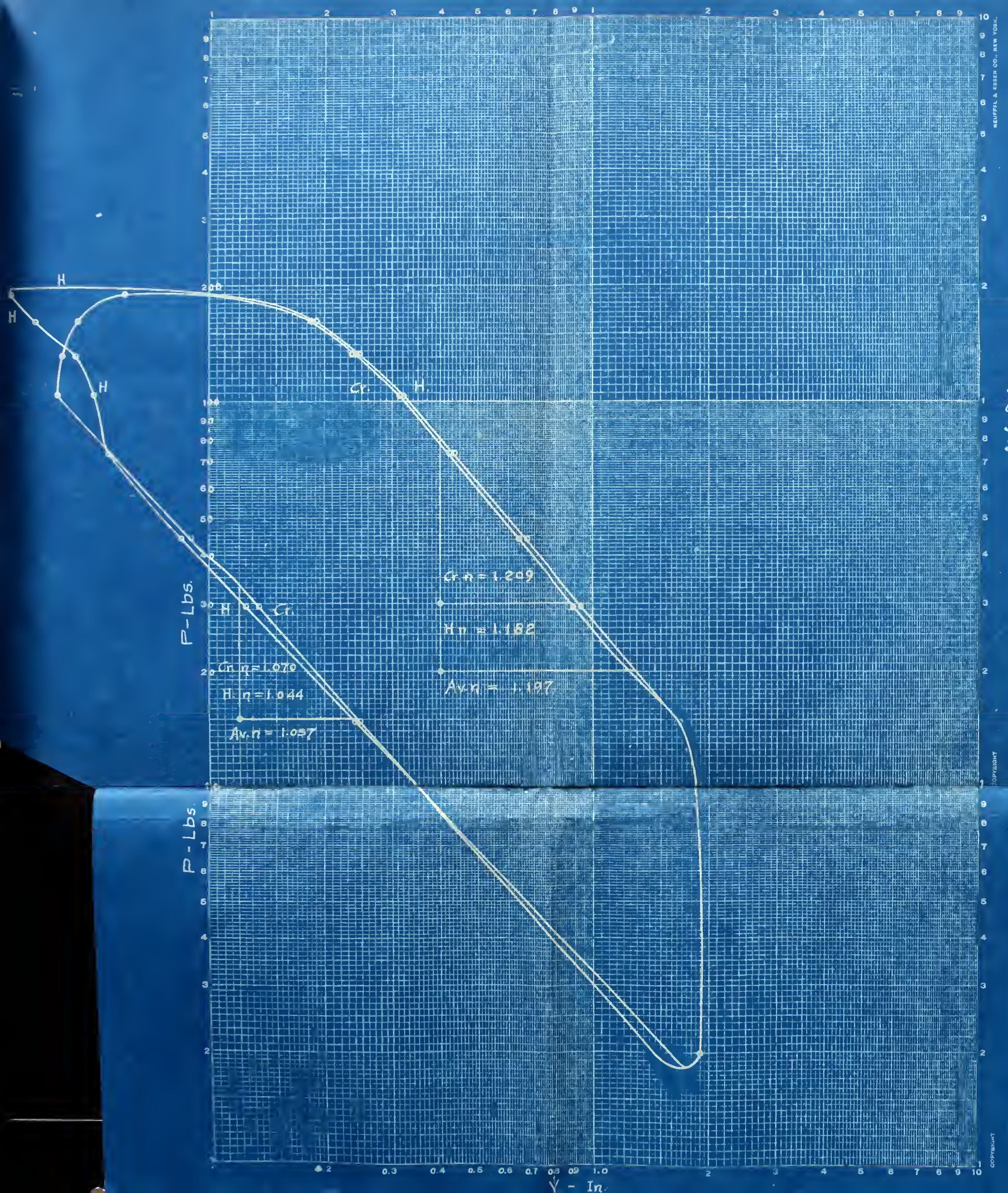

Fig. 15
Stumpf 23½" x 31½"
Uni-directional flow
Steam engine - using
Superheated Steam

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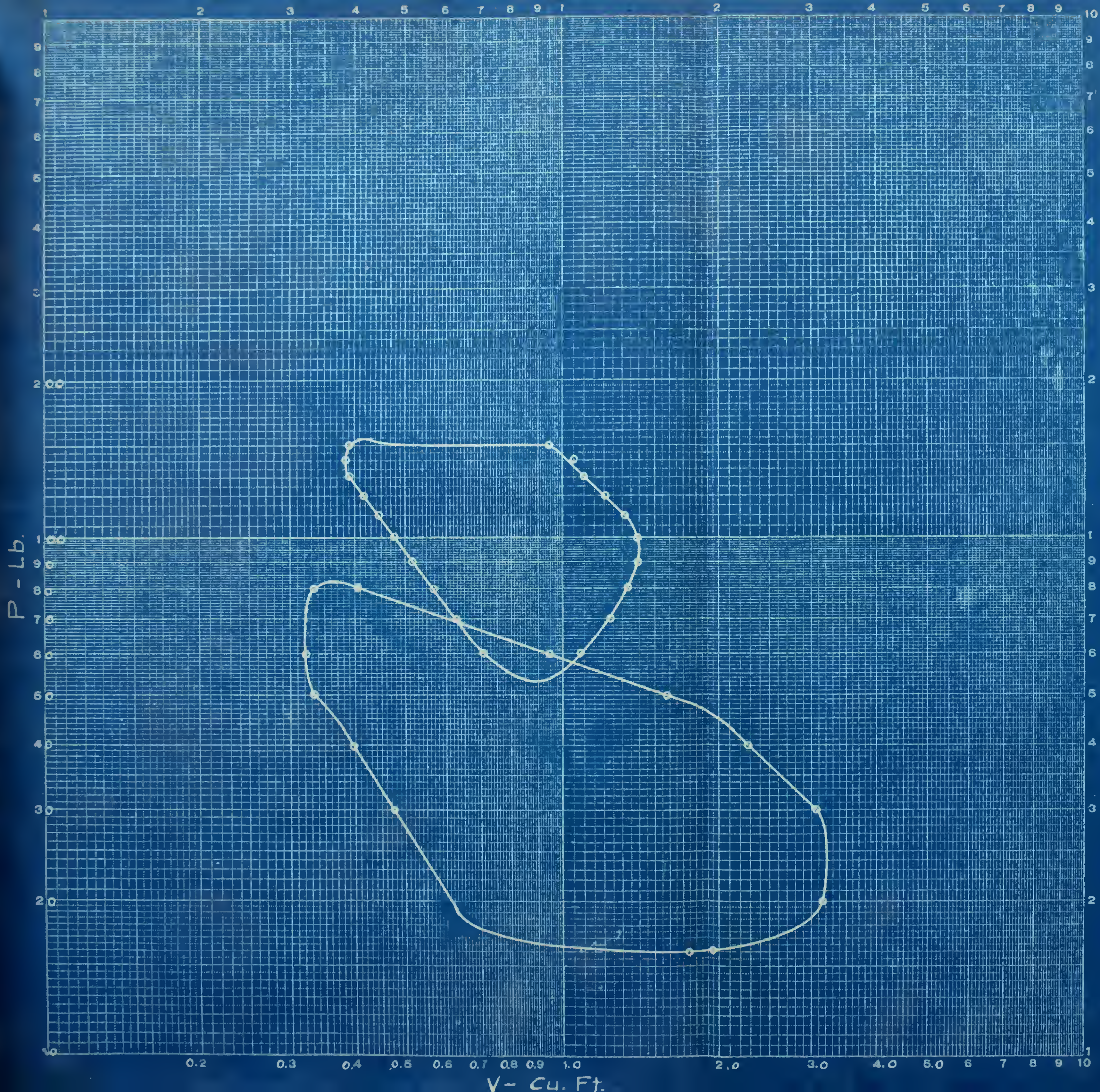

Fig. 16
Westinghouse Single-Acting
13" x 22" x 13"
Steam Engine

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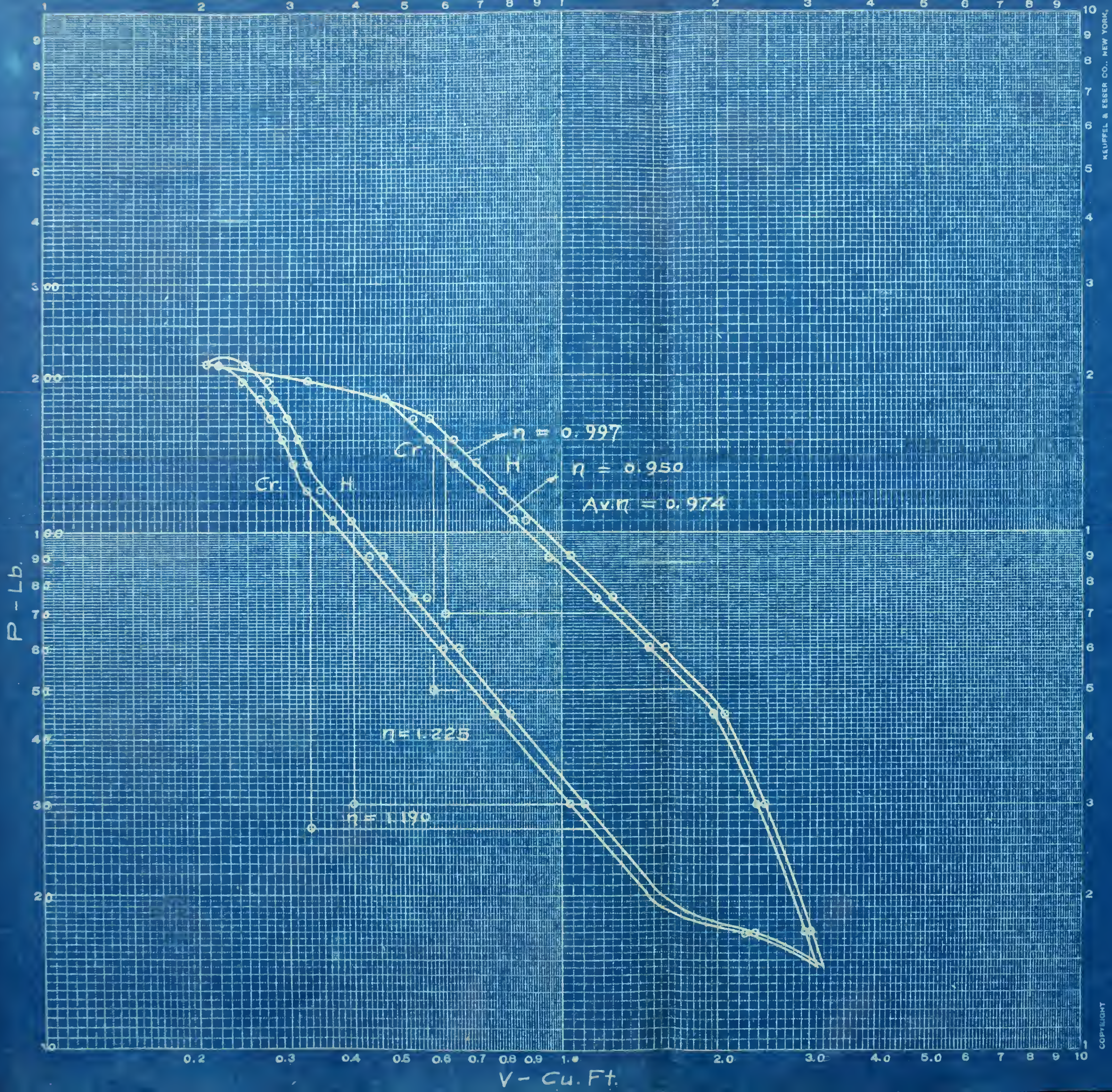

Fig. 17
Purde 16" x 24"
Superheated Steam
Locomotive

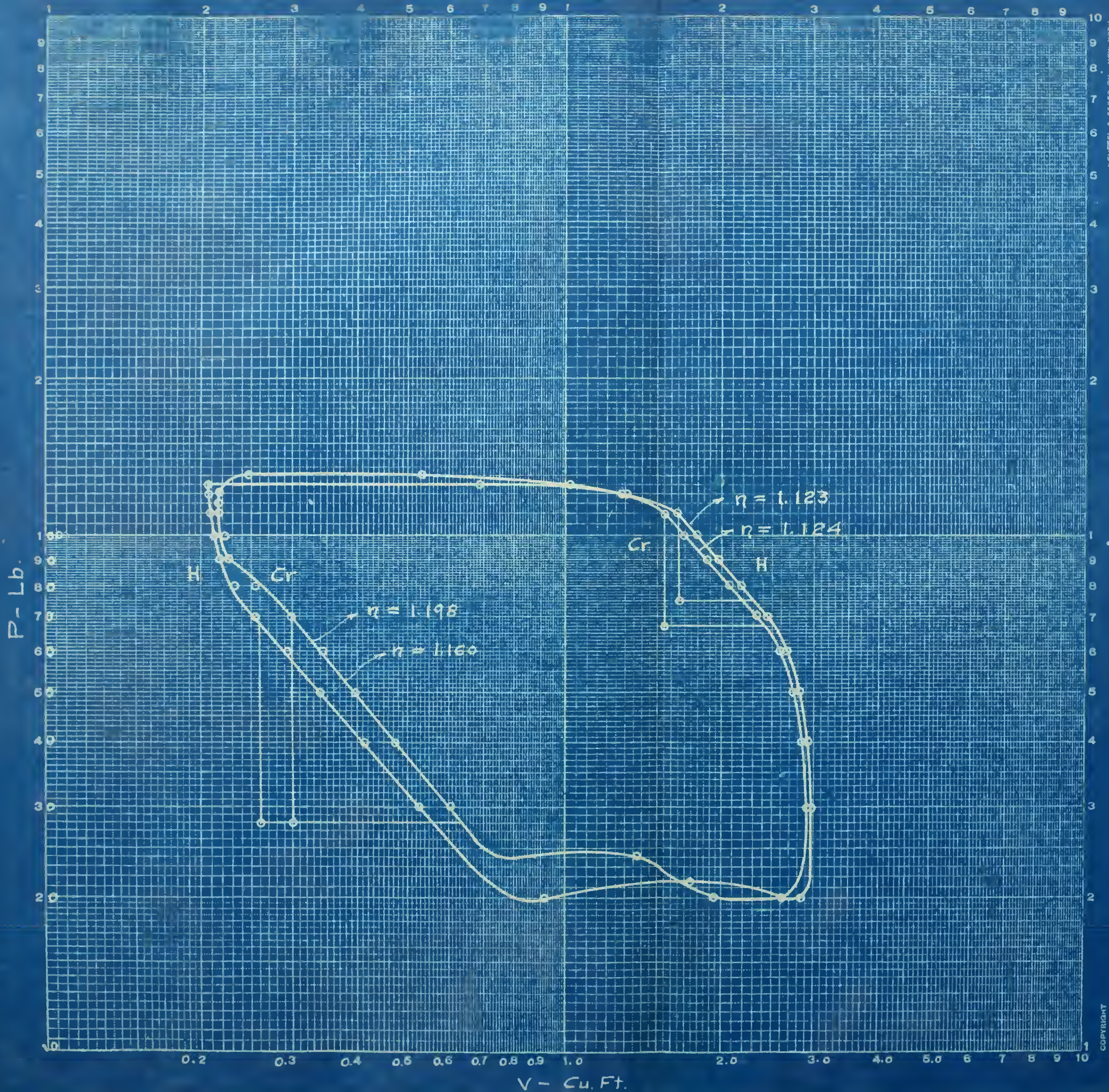

Fig. 19
Purdue 16" x 24"
Superheated Steam
Locomotive.

P - Lb.

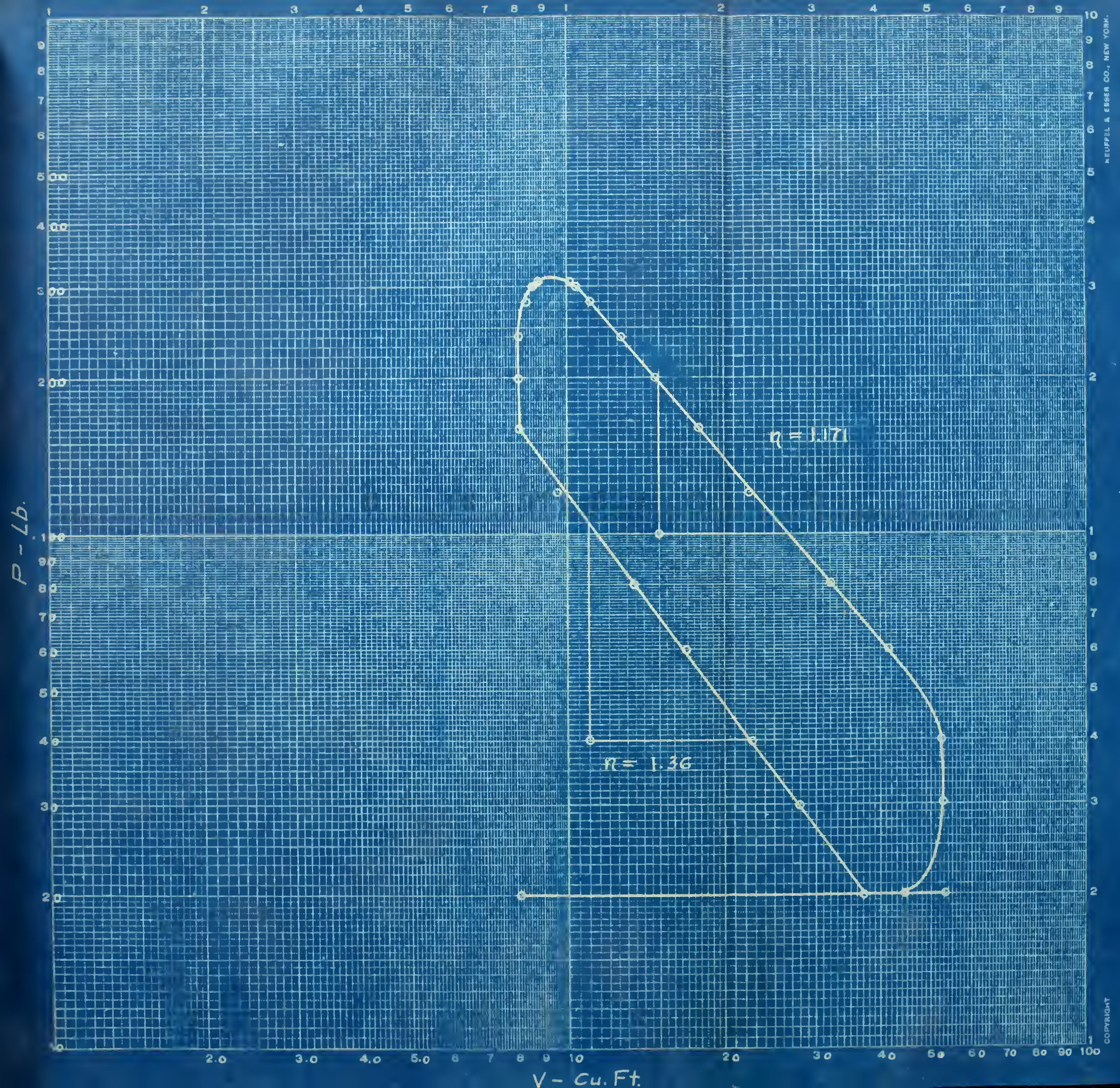

Fig. 20
Tod 42" x 60" Gas Engine
Blast Furnace

221

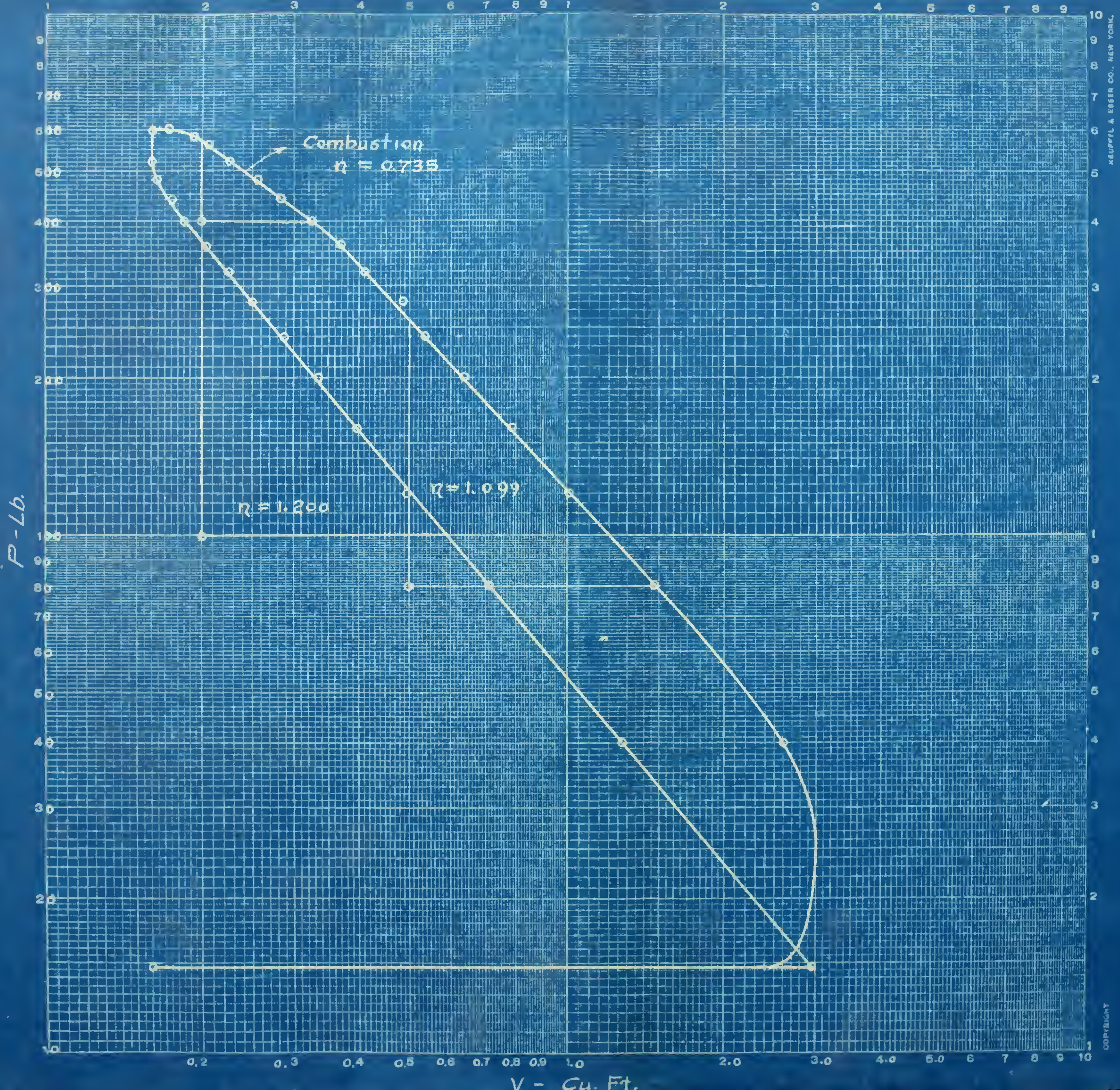

Fig. 21
Diesel 16" x 24"
Oil Engine
Crude Oil

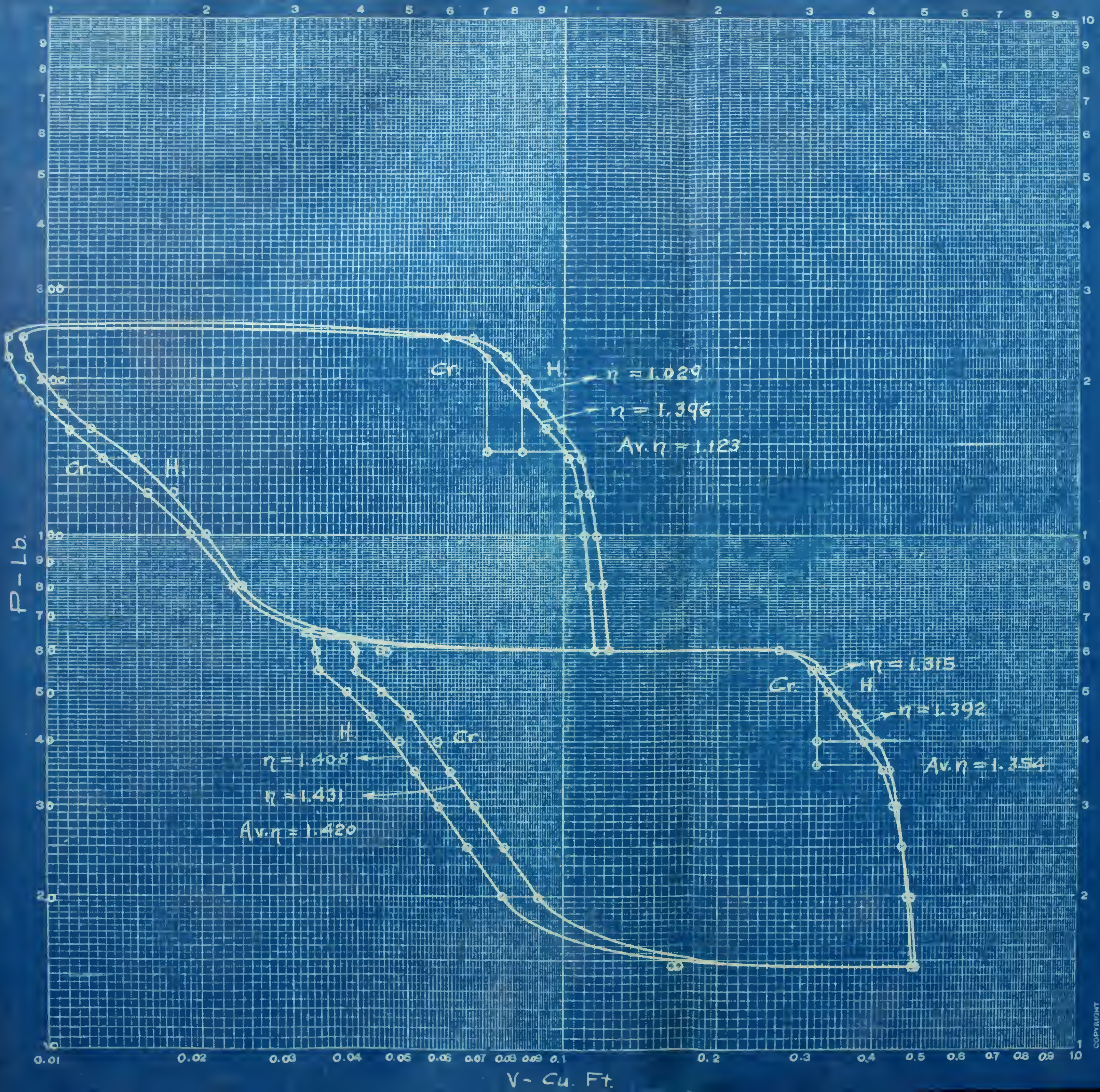

Fig. 22.
Porter Air Locomotive
Compound 5" x 10" x 10"

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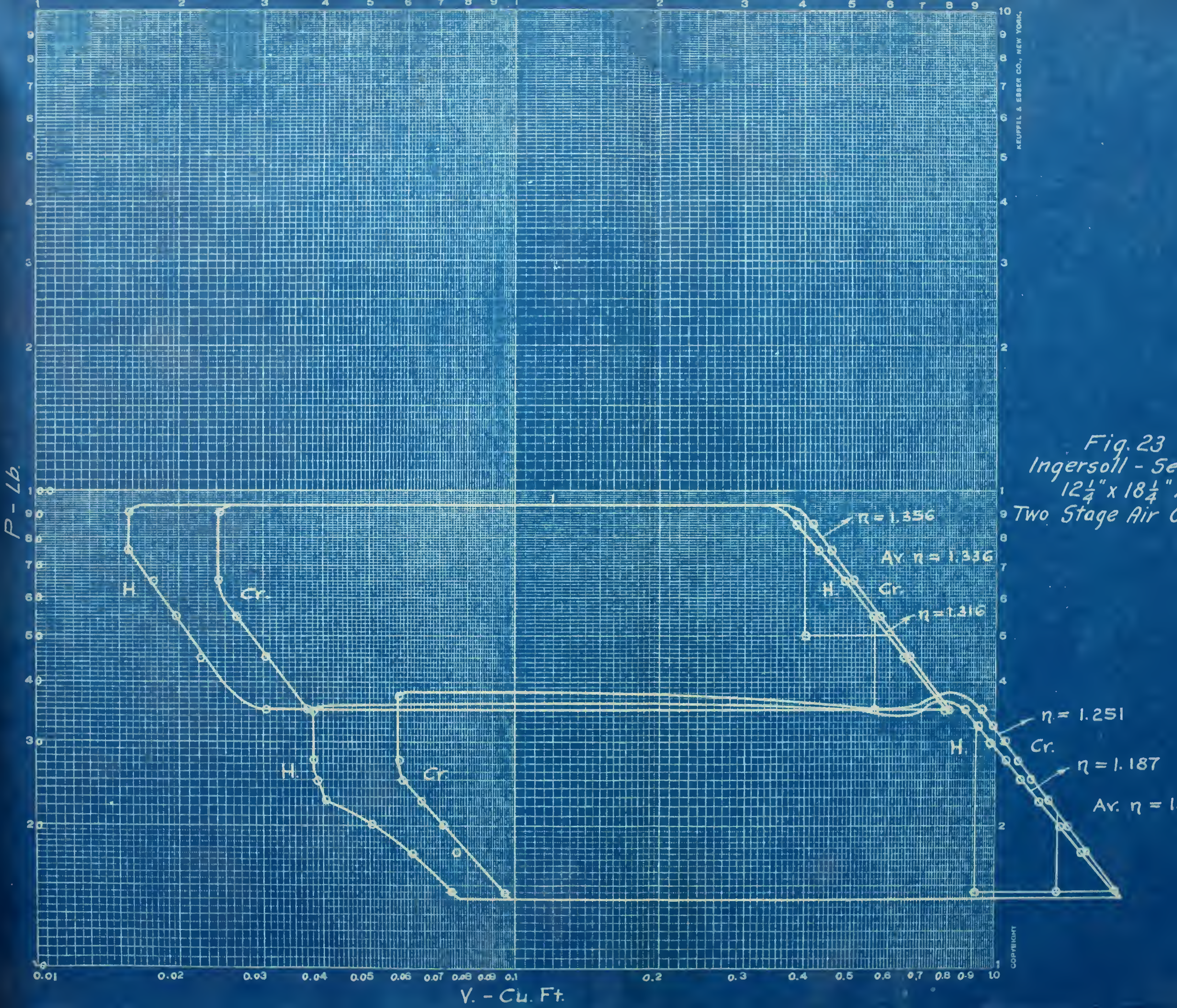


Fig. 23
Ingersoll - Seargent
$12\frac{1}{4}$" x $18\frac{1}{4}$" x 12"
Two Stage Air Compressor

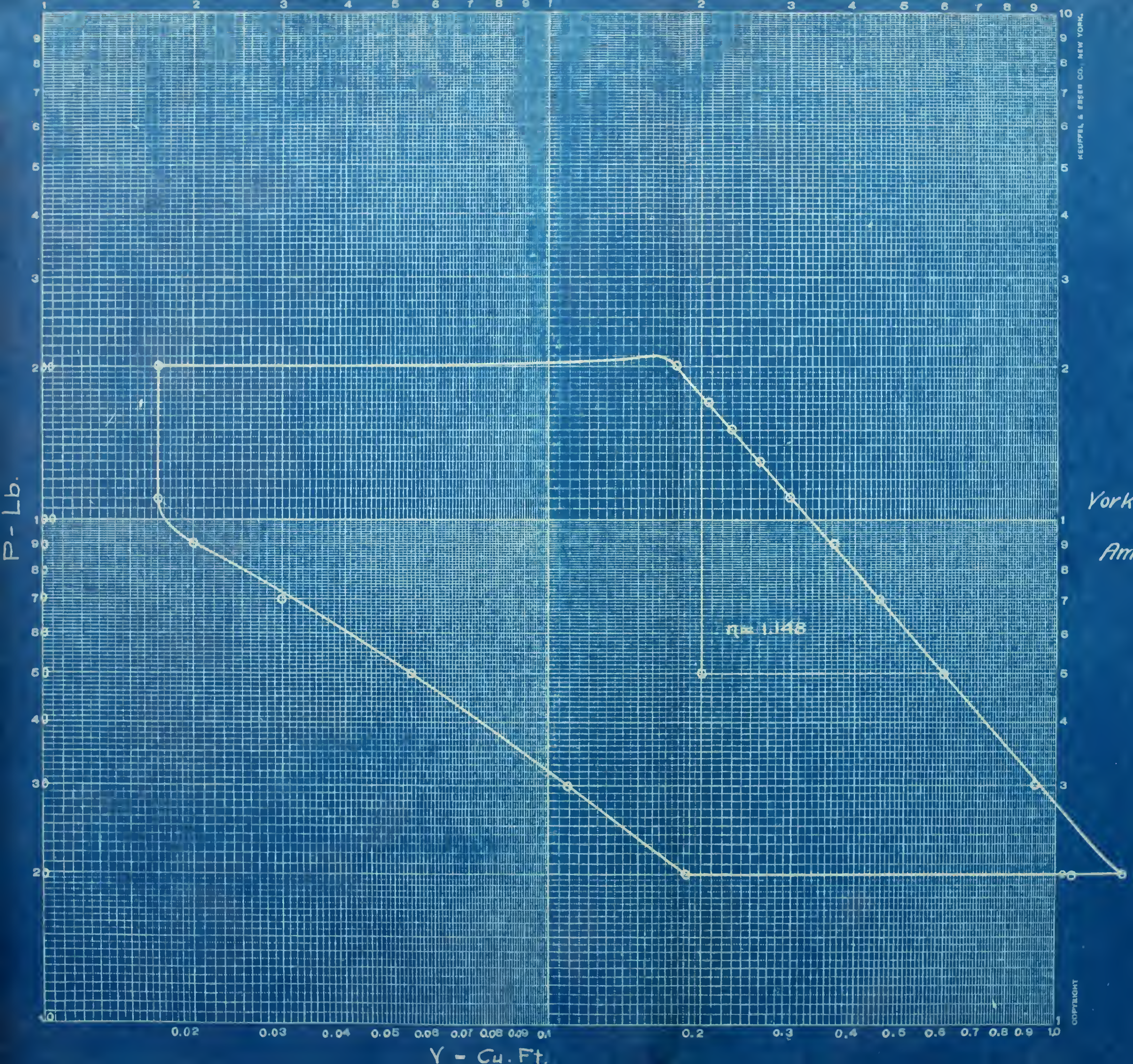


Fig. 24
York Single Acting
$12\frac{1}{2}$" x 18"
Ammonia Compressor

The writer's method of analysis is founded mainly on this fact: free from certain abnormal influences, expansion, or compression, of elastic media, takes place in cylinders according to the law PV^n = C.

a. Examples of Logarithmic Diagrams from Various Types of Engines.

The logarithmic diagrams, obtained by plotting the indicator diagrams as described in Chap. III, are of a distinctly different shape from either the PV-diagram or the temperature-entropy diagram. While the various typical forms of PV-diagrams assume rather different shapes after plotting, yet the resultant figures retain in a general way the peculiar characteristics of each PV type, except that these peculiarities are exaggerated.

Various typical PV-diagrams and the derived logarithmic diagrams are given in Figs. 11-24. They include examples from many types of engines using steam, gas, air, and ammonia as the active media. The values of n for the curves are given in each figure.

The types of steam engines represented include Corliss, high speed, auxiliary cut-off, poppet valve, single acting, and locomotive engines.

The gas engine diagrams include two four-cycle types, one using blast furnace gas, and the other a Diesel engine using crude petroleum.

The air diagrams contain one set from a compound air locomotive, and one set from a two-stage air compressor.

These figures show most of the typical forms of diagrams that are obtained in practice. At first sight the logarithmic diagrams all look distorted, but, after the meanings of the different lines become clear, they begin to seem as natural as the PV-diagram to which we are so well accustomed by long use.

These logarithmic diagrams show how clearly the law, $PV^n = C$, holds in actual curves from a great variety of engines using different media.

2. CASES WHERE THE LAW, $PV^n = C$, DOES NOT HOLD.

The curves of expansion and compression, obtained from PV-diagrams, do not always follow the law, $PV^n = C$. This fact is due to several causes, some of which have been definitely determined. These causes will be treated separately.

a. Wrong Clearance, or wrong Location of the Zero line of Pressure

The law, $PV^n = C$, is only true when P and V are measured in absolute units. The clearance must be accurately determined in order to give absolute values of V. The scale of the spring, for the PV diagram analyzed, must be known, and the atmospheric line drawn by the indicator, in order to locate the zero line of pressure. The units used for P or V may be of any denomination, but they must be measured from the zero of P and V.

When a curve, $PV^n = C$, is plotted on logarithmic paper, the resultant curve is a straight line. The value of n, as already explained, is the slope of this line, measured from any two points. When the values of P and V are not absolute values, this curve is no longer a straight line, but becomes a curve of the second degree. The value of n, being the slope obtained from two points on this curve, is no longer constant for all parts of the curve, but varies from point to point.

Therefore, when PV-diagrams are transformed to logarithmic diagrams, the values of both P and V must be measured in absolute units. When these values are not in absolute units, the resulting curve is not of the form, $PV^n = C$, and therefore is not a straight line on logarithmic paper. The shape of the curve obtained, when the values of V alone are not in absolute units, is given in Fig. 29.

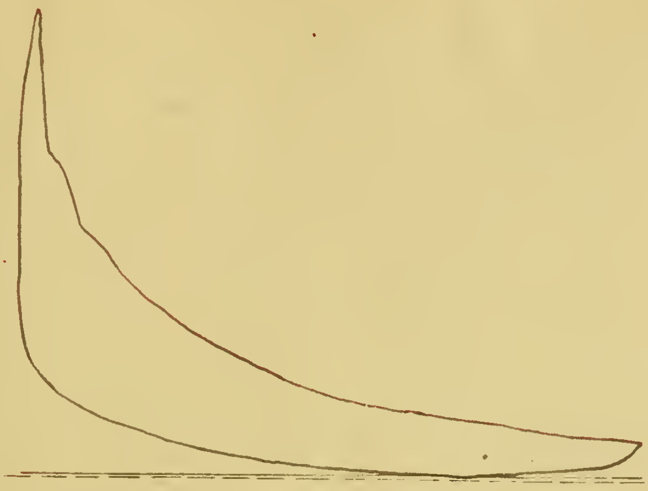
b. Leakage.

The law applies only to cases where the weight of the working medium remains constant during any expansion or compression. When this weight changes materially, either by leakage into, or out from, the cylinder containing the medium, the resulting expansion or compression no longer follows the law, but becomes a curve on logarithmic paper. This is very clearly shown in the curves from engines where large leaks were known to exist.

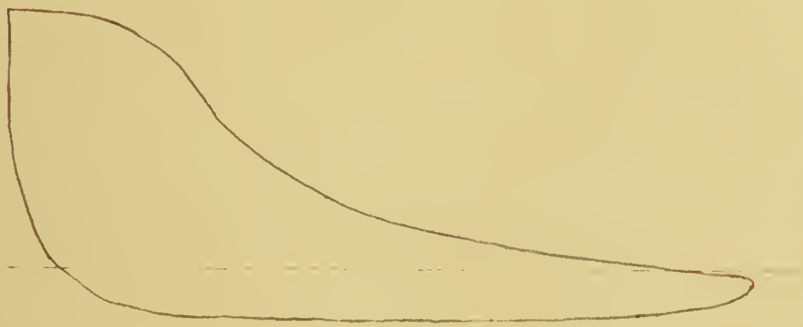
The first case, shown in Fig. 25, occurred in a 10 in. X 19 in. gas engine, intended for producer gas, but using illuminating gas at high compression. The piston, a single-acting trunk type, allowed a large leak, clearly detected by the noise of escaping gas, at the beginning of the stroke. Both the compression and expansion curves show the effect of this leak in a clear manner when transferred to the logarithmic form. After that portion of the stroke was reached where no sound of leakage was heard, the two curves became straight lines. This indicated very clearly that the effect of leakage, if appreciable, may be detected in the form of the curves of the logarithmic diagram.

The second case, shown in Fig. 26, is from a 14 in. X 35 in. Corliss engine. The knowledge of the leaky condition of the piston and valves came from the engineer in charge, but was told to the writer by Professor O. A. Leutwiler, who took the diagram shown.

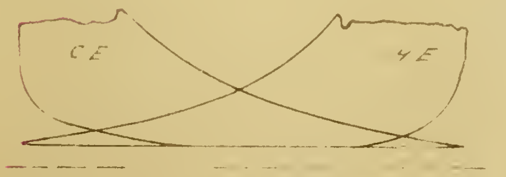
The expansion and compression lines indicate by their shape at the upper ends, a large leak from the cylinder, either by the piston, or through the exhaust valve. The lines also show, by the raising of the curves at the lower ends, a considerable addition to the steam in the cylinder during both expansion and compression. This steam could only come from a large leak in the steam valve. The seven other diagrams taken from this same engine all showed the effect of



Scale 190#
Fig. 25



Scale 40#
Fig 26



Scale 160#
Fig 27

725

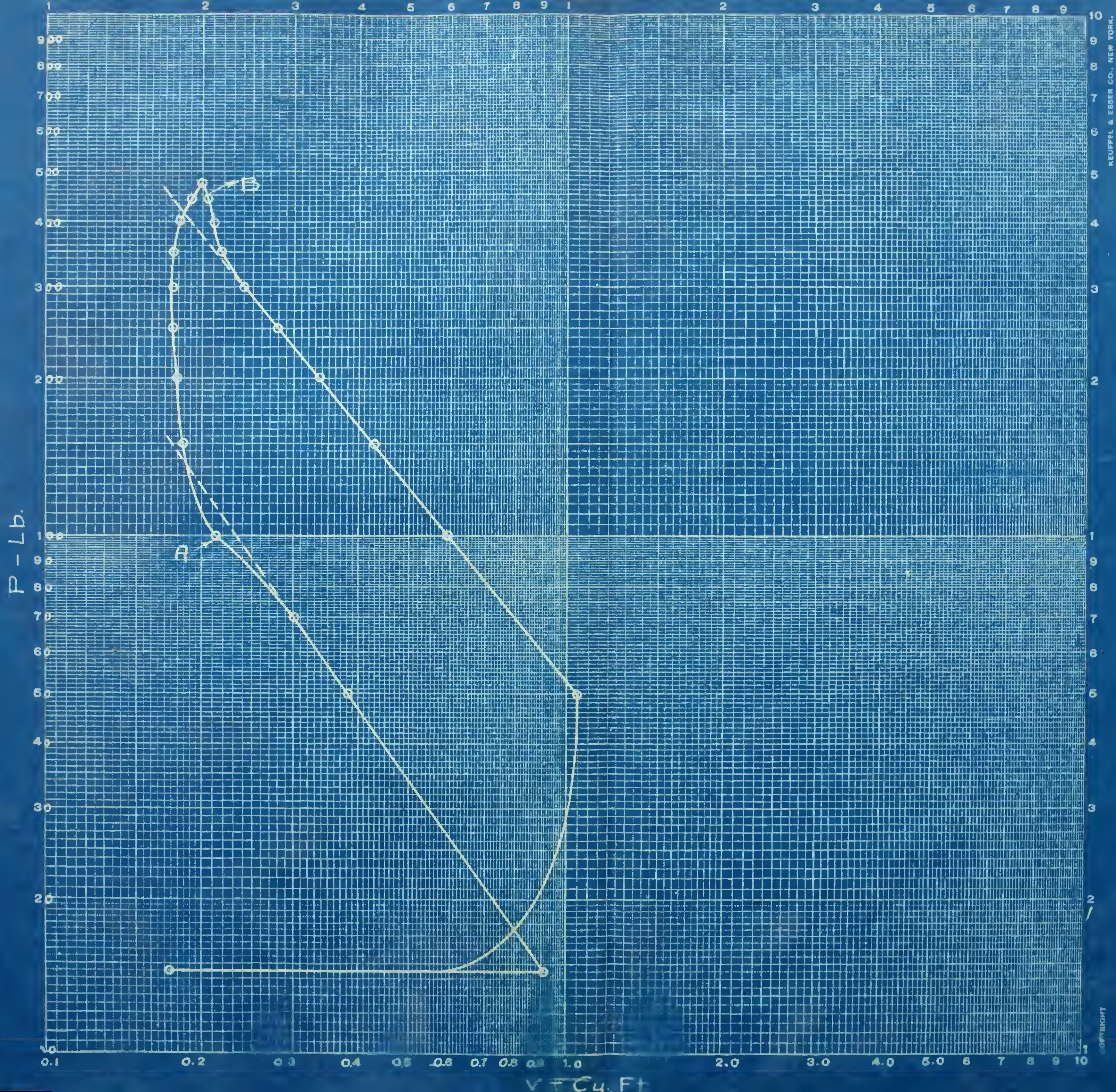

P - Lb.
V - Cu. Ft.
A
B
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Fig. 25

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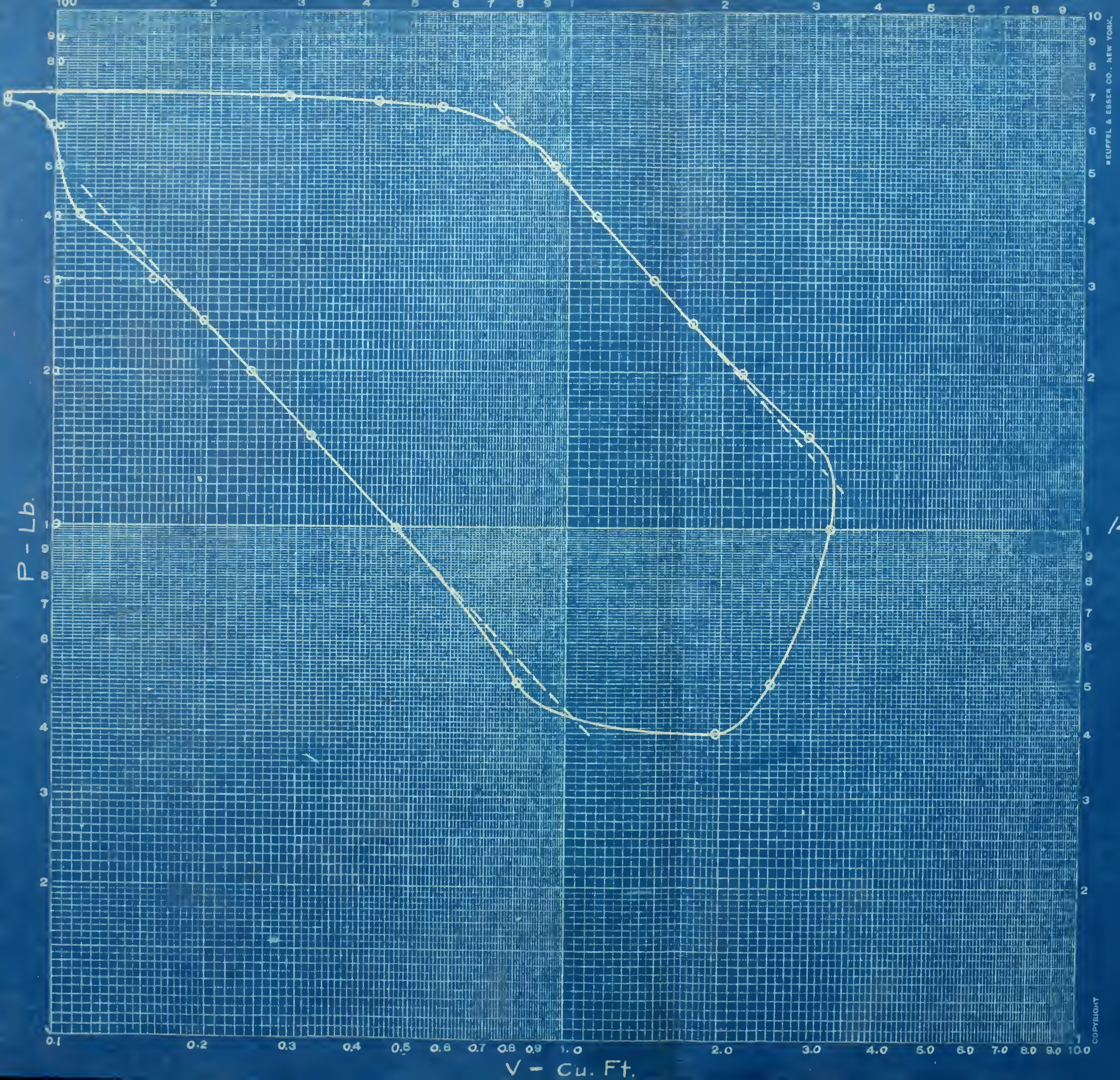

Fig. 26
14" x 35" Corliss Engine

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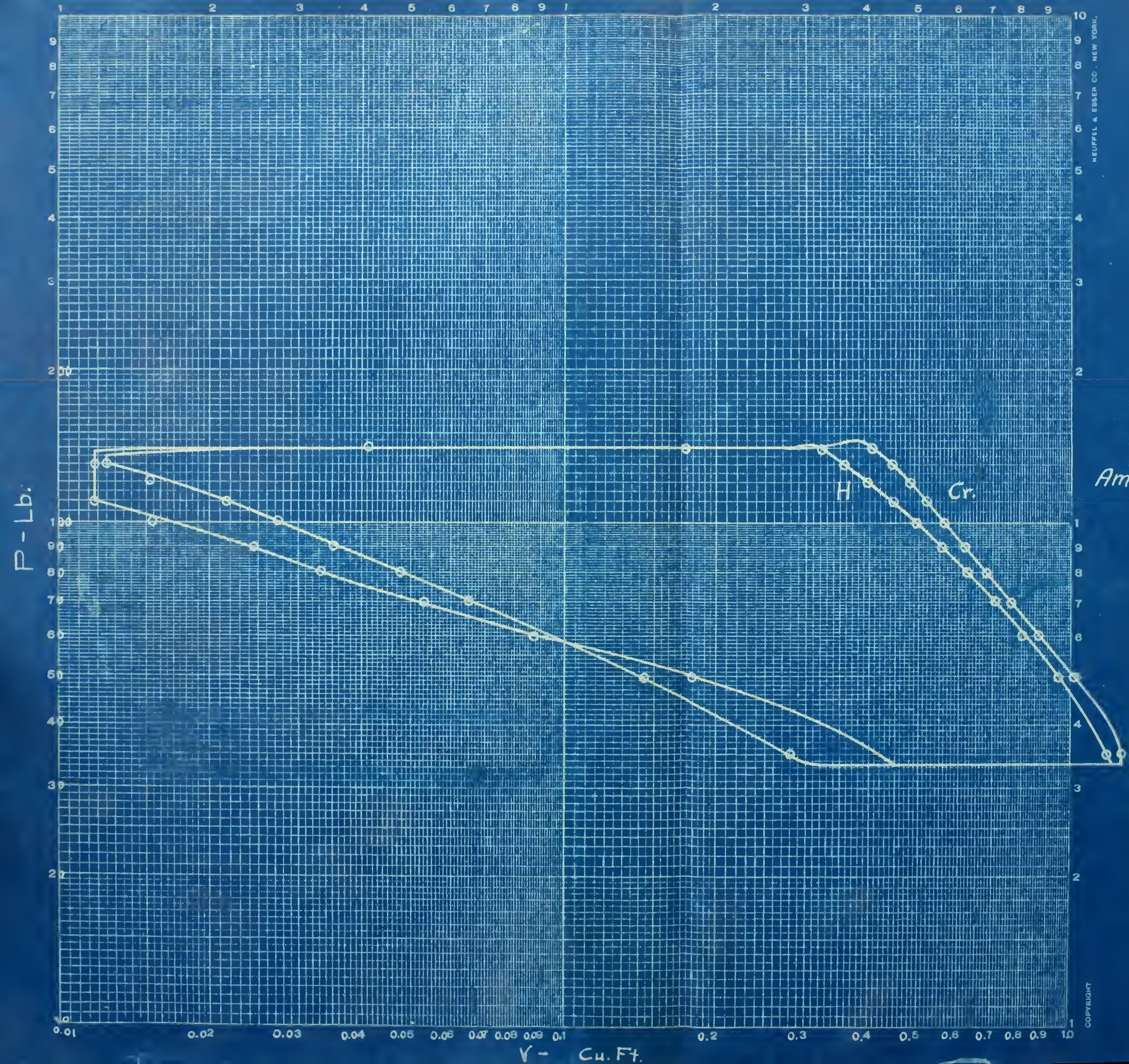

Fig. 27
Ammonia Compressor
11¼" x 22"

leakage in a similar manner.

The third case of known leakage, shown in Fig. 27, is from an 11 1/4 in. X 22 in. double-acting ammonia compressor. This cylinder was known to be in very bad condition as regards wear and leakage of piston and valves. The re-expansion curves, by the enormous amount of re-expansion shown, indicate large leakage into the cylinder during this operation. The lower part of the compression curve, by rising, indicates leakage into the cylinder, either past the piston or through the discharge valves. The upper part of these curves indicates leakage from the cylinder, either past the piston, or through the suction valves.

These three examples show abnormal conditions which are comparatively rare. The distinguishing of one type of leak from another is treated in Chap. VII.

c. Low speed in Steam Engines.

Very low speeds of rotation, together with very low piston speeds, will cause the compression curve to deviate from the law $PV^n = C$. The most common cases of this effect are seen in the "hook", or excessive condensation, near the upper end of compression curves. These "hooks" are found almost altogether in small engines having very low piston speeds, and in larger engines with small clearance, having very low rotational speeds, as in pumping engines.

Diagrams containing "hooks" in the compression curves have been published by Professor Dwelshauvers-Dery from the experimental engine at Liege. This is an example of a very low speed in a small engine. The size was 12 in. X 24 in., and the speed from 30 to 60 R.P.M. That this hook was caused by low speed and consequent excessive condensation, was proved to the satisfaction of the writer in the case of the engine described in Appendix I. This engine was also 12 in. X 24 in., but was operated at from 90 to 150 R.P.M. In no case

was a hook in the compression curve obtained during the tests, although some 1600 diagrams were taken. A set of diagrams from these tests is shown in Fig. 2, and shows no sign of a hook. On one occasion however, the diagram shown in Fig. 28 was obtained. This was taken just after the engine was started from cold, and had been brought up to a speed of 120 R.P.M. In conjunction with the other diagrams obtained from this engine in regular operation, this hook is believed to be due to excessive condensation while the cylinder was comparatively cold.

The writer has taken diagrams with the hook present in the compression curves from only two other engines; one was an 8 1/2 in. X 12 in., running at 110 R.P.M., and the other 5 in. X 6 in., running 140 R.P.M.

This excessive condensation at the end of compression, or near the dead center, is due to the fact that the deleterious surface effect of the cylinder walls is so enormous compared with the weight and volume of steam present at this point.

The three causes treated above are believed to be the important conditions that cause the curves in practice to depart materially from the law $PV^{\underline{n}} = C$. Sometimes only one of these conditions is present, while in other cases, a combination of these may influence the resulting curves. The separation of these conditions, by their effect on the curves, is treated in Chap. VII.

3. RATIONAL METHOD OF APPROXIMATING CLEARANCE.

a. Method of Graphical Trial and Error.

In the cases of the great majority of diagrams analyzed, the curves became straight lines in the logarithmic diagram, showing that the law $PV^{\underline{n}} = C$ was applicable, or in other words, that $\underline{n}$ was a constant for one curve. The clearances furnished with the diagrams analyzed had all been carefully found by the displacement method.

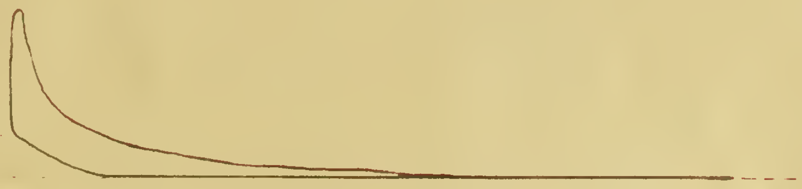

Fig. 28
Taken while starting from cold.

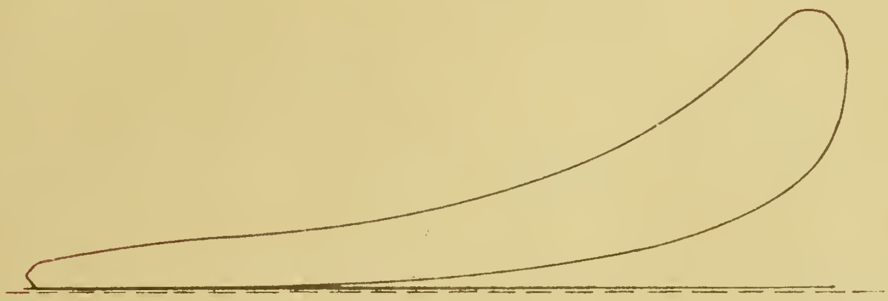

Fig. 29
Scale 160#

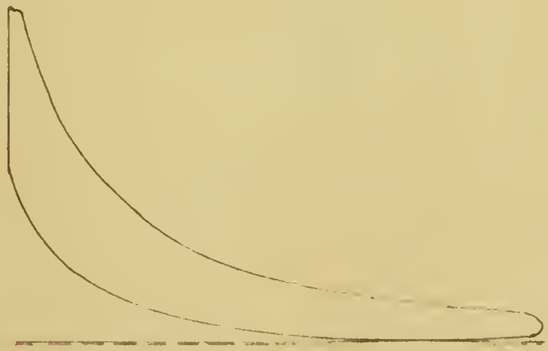

Fig. 30
Scale 180#

It was desired to see what forms the lines assumed when the clearance was taken larger or smaller than the measured quantity. The diagram shown in Fig. 29, taken from a 42 in. X 60 in. gas engine, was used for this purpose. The true clearance, carefully measured as 18.0%, was used in the full diagram. Trials were made with clearances assumed as 14.0, 16.0, 20.0, and 22.0 per cent of the piston displacement. With the clearance of 18.0%, the curves became almost perfectly straight lines. With the values of clearance less than 18.0, it is seen that the lines become bent to the left, and with values over 18.0, the lines become bent to the right. Hence the straight line for the value 18.0 is the transition between the family of curves bending to the left representing a clearance smaller than the real value, and the family of curves bending to the right representing a clearance larger than the real value.

The practical significance of this fact is that we have now available an entirely rational method of approximating the clearance of any cylinder, using an elastic medium, which has, as a part of the cycle of operation, an expansion or a compression. This method is based on the fact, already mentioned, that in practice all elastic media, except in certain exceptional conditions, obey substantially the law $PV^n = C$, when subject to change of state, and therefore become straight lines in the logarithmic diagram.

This method requires only the scale of the indicator spring to be known, and the atmospheric line to be drawn, in order to locate the zero line of pressure. The order of procedure necessary to make a trial, and the degree of accuracy obtained in any given case, is shown in detail in Chap. VII for a 25 5/8 in. X 37 5/8 in. gas engine. All that is necessary is to assume different values of clearance, and plot the logarithmic diagram for each assumed value. The straight line position of the curves is found by trial and error as lying between the

P - Lb.

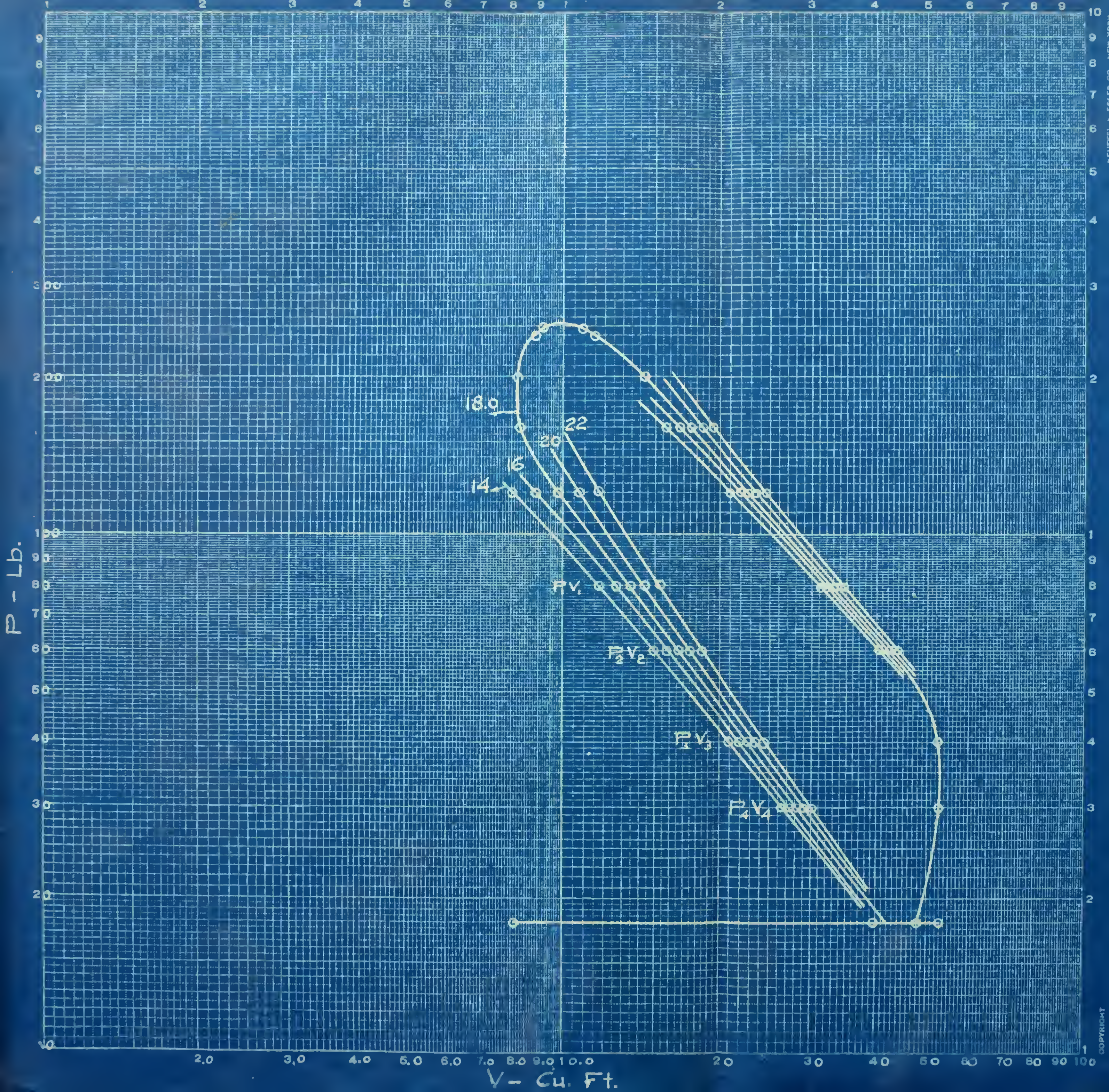

Fig. 29
Tod 42" x 60" Gas Engine
Blast Furnace Gas

two diverging families of curves representing too small and too large clearance.

It follows also, from the curves, shown by Fig. 29, that, the clearance being known, the scale of the spring used may be obtained in the same manner if the atmospheric line is given.

b. Mathematical Interpretation of the Clearance Trial Method.

The results obtained from the graphical method of trial and error may also be accomplished by the use of the purely mathematical process upon which the method depends.

The law $PV^n = C$ holds, thus n is a constant for any part of the curve. When the wrong clearance is used, the law $PV^n = C$ does not hold, and n varies from point to point. In the graphical method, trials of various values of clearance are made, until the curves become approximately straight lines. This resultant straight line is therefore the law $PV^n = C$, in which n is a constant for all parts of the curve. The one condition necessary to be fulfilled, therefore, is that n be constant for all parts of the curve, but not of any particular value.

To illustrate the use of the mathematical method in Fig. 29 let us assume several points, as P_1V_1, P_2V_2, P_3V_3, and P_4V_4 at various intervals on one of the curves, as on the compression curve at the clearance value 14.0. It is desirable for convenience to locate the points at about equal intervals, as shown. The law $PV^n = C$ is assumed to hold. Then for two points, P_1V_1 and P_2V_2, called group a, we have

$$P_1V_1^n = C. \; . \; . \; . \; . \; . \; . \; . \; . \; . \; (1)$$

$$P_2V_2^n = C. \; . \; . \; . \; . \; . \; . \; . \; . \; . \; (2)$$

Equating these, we obtain

$$P_2V_2^n = P_1V_1^n. \; . \; . \; . \; . \; . \; . \; . \; (3)$$

Transposing and dividing,

$$(V_2/V_1)^{\underline{n}} = P_1/P_2 \text{. . . . .} \quad (4)$$

Taking the logarithm of both sides,

$$n(\log V_2 - \log V_1) = \log P_1 - \log P_2 \text{. . .} \quad (5)$$

Whence

$$\underline{n}_a = \frac{\log P_1 - \log P_2}{\log V_2 - \log V_1} \text{. . . . .} \quad (6)$$

In the same manner for the points P_3V_3 and P_4V_4, called group $\underline{b}$, we obtain

$$\underline{n}_b = \frac{\log P_3 - \log P_4}{\log V_4 - \log V_3} \text{. . . . .} \quad (7)$$

For the correct value of clearance, this condition must be fulfilled

$$\underline{n}_a = \underline{n}_b \text{. . . . .} \quad (8)$$

The values of the logarithms of the coordinates of all the points are then found, and the values of $\underline{n}_a$ and $\underline{n}_b$ computed. If the points are located in the order shown, then with too small a clearance $\underline{n}_a$ is lower in value than $\underline{n}_b$. A larger value of clearance is then assumed, the operation being merely to add a constant number to the values of V_1, V_2, V_3, and V_4. The process is repeated until the value of $\underline{n}_a$ becomes practically equal to $\underline{n}_b$. When the clearance is assumed too large, $\underline{n}_a$ becomes higher in value than $\underline{n}_b$, in dicating that the true value has been passed.

The trial by the mathematical method is not as accurate nor as short as the graphical method. It is not as accurate because the points assumed may not be representative. When this is the case, the graphical method allows judgment to be exercised in selecting the straight line position, thereby eliminating irregularity of points.

The question arises as to whether the shape of the lines due to wrong clearance can be distinguished from the shape due to leakage or "hooks", on the logarithmic diagram. This case is treated in Chap. VII.

The curve of the PV-diagram nearest the clearance space, or

the compression curve in Fig. 29, is generally the better guide in the graphical trials. This is well shown in Fig. 29. A given difference in the values of clearance used for trial causes more horizontal variation in the position of the compression curve than in the expansion curve. This fact allows closer locations of the straight line region transition to be made from the compression curve than from the expansion curve.

4. RATIONAL METHOD OF OBTAINING THE STROKE POSITION OF CYCLIC EVENTS.

It is often desirable to know at what part of the stroke the cyclic events occur. This knowledge can be best obtained from the PV-diagram. For ordinary purposes these events can be closely located on the PV-diagrams in most cases. Thus on a diagram from a Corliss engine, cut-off may generally be located to within 1/16 inch. The actual beginning of compression, however, can never be accurately on the PV-diagram. Compression itself begins after the exhaust valve, in closing, has acquired enough seal to prevent leakage. This point is generally at least 5 lb. above the back pressure. The point of no leakage cannot be seen because the curve of compression, and the curve during the time the valve has insufficient seal, are of the same direction of curvature, and are not reverse curves as in the general case of admission and expansion.

The fact that expansion and compression of a constant weight of medium take place according to the law $PV^n = C$, thus becoming straight lines on logarithmic paper, enables us to locate cyclic events very closely, even in cases where they cannot be detected at all in the PV-diagram.

An example is shown in Fig. 17. These are locomotive diagrams taken at short cut-off and high speed. The events at cut-off, release, compression, and lead are very difficult to locate on such diagrams. These events are located on the logarithmic diagram by

noting when the expansion and compression curves become straight, indicating a constant weight of steam. A sufficient number of points are plotted to show clearly the direction of the diagram near the events desired. Thus these events, even though obscure in the PV-diagram, may be located to within about 1/16 inch in the logarithmic diagram, or to about 1/25 inch on the PV-diagram. This is almost as accurate as is the diagram itself.

The use of this method has one great advantage in that it eliminates largely the variable element of personal judgment. The writer has seen PV-diagrams where two persons have located an event, such as cut-off, 1/8 inch apart, each location being the best judgment of the person doing the work. The logarithmic diagram will at all times give closer locations of events for these reasons than will the PV-diagram.

5. A RATIONAL METHOD OF COMPUTING STEAM CONSUMPTION FROM THE INDICATOR DIAGRAM.

The most reliable method of determining the steam consumption of any engine is to measure or weigh the water directly, preferably by a surface condenser. This necessitates an elaborate test, which disturbs the routine of the plant tested, and is very costly for long tests of large engines. The objections to tests of this kind are many; and a few of them will be stated. When the boiler feed is measured, both the engine and the boilers serving it have to be entirely disconnected from other units. This sometimes necessitates shutting down the rest of the plant. Almost all tests, where there is more than one unit, necessitate changes in heavy and permanent piping. Boilers are apt to leak in service, and the measurements of these leaks are unsatisfactory. One serious objection to long time tests is that the different rates of steam consumption cannot be separated.

The ideal method of testing a steam engine would be a method

analogous to that used with electrical machines, that is, to measure instantaneous rates of consumption instead of the water consumed over a long time.

On account of the cost and difficulties of making a test, many engineers have devised methods of approximating the steam consumption without actually measuring it.

When indicator diagrams were first obtained, the loss from initial condensation, or the existence of the "missing quantity", was not suspected. The opinion, therefore, was that the consumption could be measured from the steam shown by the diagram at cut-off. After Isherwood made his tests on the U. S. S. "Michigan", the existence of this "missing quantity" was revealed. The very great difference, in the proportion that this "missing quantity" bears to the total weight of mixture present, either at cut-off or during expansion, that is found in different types and sizes of engines has prevented any reliable determinations of the actual steam consumption by this method. The steam consumption computed from the diagram, when using saturated steam, is generally from 10% to 50% below the actual consumption.

The devising of an accurate method of measuring the actual weight of steam consumed from the diagram has therefore been regarded as impossible. Thurston states that "the steam or water consumption of an engine cannot be exactly ascertained by the use of the indicator" for the reasons mentioned. Most other writers on the subject have expressed similar views.

Methods of computing the weight of the "missing quantity", from the physical dimensions, type, and speed of the engine considered, as shown by the results of large numbers of tests, have been devised and published by Escher, Cotterill, Heck and others. The results obtained by these methods have not been uniform, and do not agree closely enough with the test results to be used with confidence.

Professor Heck states that the consumption computed by the use of his formula should ordinarily not show more than 10% difference from the test results.

The significance of the facts disclosed in Chap. V, that the proportion of this variable and unmeasured condensed steam, the "missing quantity", controlled absolutely, in any one engine, the value of n for the resulting expansion curve, furnishes us with an entirely rational method of computing the actual weight of steam and water present from the indicator diagram.

The relation of Q and n, for the same engine, has been shown to be very definite under the same conditions of pressures and speed. This relation, or dependence of n upon Q, however, is not seriously affected by changes of pressure and speed within the limits of the tests.

The ordinary speeds of similar types of engines, 70 to 120 R.P.M., do not affect the relation seriously enough to be taken into account when examining such cases. This is because of the two-fold character of speed, in its influence upon the action of the cylinder walls.

The engine experimented upon was operated at 120 R.P.M., and had a stroke of 2 feet. Other engines of this class run at speeds as low as 70 R.P.M., but have strokes of 5 or 6 feet. Cylinder condensation is not dependent upon rotative speed alone, but is also influenced by the piston speed, as determined by the length of the stroke. On account of different lengths of stroke, different engines cannot be compared on the basis of rotative speed. Thus while the small engine tested has a rotative speed of 120 R.P.M., its piston speed is only 480 feet per minute. In large engines, while the rotative speed may be only 70 R.P.M., yet with strokes of 6 feet, the piston speed

is 840 feet per minute. What the small engine gained by higher rotative speed, the large engine made up by higher piston speed.

After taking into account the two parts of which speed is composed, it is found that the speeds of stationary engines of the type tested are in substantially the same range. On account of this fact only the results of the tests run at 120 R.P.M. have been used in the applications made at present.

The relation of Q and n has been found to be practically independent of cylinder size. This statement is true for unjacketed cylinders and for pressures in the range examined. This is shown in the following discussion for saturated steam.

Cast iron is universally used for steam cylinders. The greatest source of loss in the cylinder is due primarily to the use of a metallic structure from practical considerations. The skin surface of this metal, which is a fairly good conductor, must be heated once every cycle from the temperature acquired from contact with exhaust steam, up to nearly the temperature of the admission steam, this heating being accomplished by the condensation of some of the incoming steam.

The amount of this condensation, measured as the proportion of the mixture present, varies with the size, valve design, relative roughness of the interior surface, temperature range, length of cut-off, speed, location of ports and port passages, quality of the steam supplied, and the jacketing and lagging. It can easily be seen, from the number and relative magnitude of these variables, that the computation of the weight of condensation, by means of a formula which will take these variables into account, can never be an accurate operation.

The writer, after many examinations into the cases of different types and sizes of engines, with unjacketed cylinders in good order, and with pressure limits similar to those used with the tests,

finds that, while the initial condensation is subject to the action of ten or more variables, yet the value of n resulting from a given value of Q is almost always substantially the same. A few of the applications proving this point will be found in Table 29. Here the cylinder sizes vary from 14 1/2 in. X 13 in. to 28 1/2 in. X 59 1/2 in., the speeds from 60 to 250 R.P.M., and the types include slow speed Corliss, high speed, and locomotive engines. The possibility of calculating accurately the weight of condensation in these different cases may be easily imagined.

The phenomena caused by the presence of the cylinder walls, in the class of engines discussed, have been found to be divided into two natural classes: those occurring before cut-off, and those occurring after cut-off.

The phenomena occurring before cut-off are controlled by the action of the ten or more variables already mentioned, and therefore are subject to all the variation that may occur in any individual case to be examined. For this reason any method of computing the condensation accurately is open to objection.

The phenomena occurring after cut-off are practically independent of all variables except Q, and the initial pressure and speed. Of these variables, only the value of Q and the initial pressure have proved to be of material importance in the applications made thus far.

This may be summed up by stating that the value of Q, in any particular case, is subject to the action of many important variables, but that the resulting value of n is independent of these variables, but depends only upon the value of Q, and the pressure and speed at which Q was obtained.

It appears therefore that this new method of approximating the value of Q at cut-off, from experimentally determined relations, and thus accounting for the "missing quantity", is upon much surer

ground than any method of computing condensation from the physical facts surrounding the case. It approaches the problem from the side where the phenomena occurring are practically independent of all the variables mentioned. This fact adds greatly to the accuracy obtained, and with the applications given, shows that an accurate and entirely rational method of computing the actual steam consumed from the indicator diagram alone has at last been devised.

This new method is free from several objections to which tests are open. It measures the consumption in one revolution, and is therefore practically measuring a rate instead of a quantity. The only data needed for a determination are one set of indicator diagrams, taken simultaneously, the physical constants of size and clearance, and the speed of the engine tested. No interruption of any kind in the routine of a plant is caused, and the expense incurred is not to be compared with that of an equally accurate test. It is accurate enough for almost all purposes except guarantee tests, subject to bonus and forfeit contracts. In the case of locomotives on the road, it is the only possible method of measuring the steam consumption of the main engines, due to the use of steam by the air pump, steam heat, blower, generator sets, safety valves, whistles, blow-off valves, and leaks. The same is true of marine engines, where many auxiliaries are supplied with steam from the same boilers, and exhaust into the same surface condensers. The method is especially useful for non-condensing engines, where the boiler feed measurement method is the only practical one. Steam consumption may be obtained as often as is desired instead of probably once in the engine's life.

The relations of Q and n given in Chap. V are applicable, however, only to unjacketed cylinders exhausting at very close to atmospheric pressure. When the back pressure is raised to 30 lb., for

instance, there is a new series of relations existing for the same initial pressures, due to a different temperature range in the cylinder, and consequent alteration of the phenomena occurring after cut-off. Steam jackets also alter the phenomena after cut-off, and therefore have to be examined separately for the relations of $\underline{Q}$ and $\underline{n}$.

These facts make it necessary to examine the relations of $\underline{Q}$ and $\underline{n}$ for several different classes to include existing engines. It is proposed to examine these classes in the near future.

Since the analysis rests entirely on the indicator diagram, great care must be observed in taking these diagrams. The indicator itself must be an accurate instrument in the best possible condition. The indicator connections must be short and direct. An extensive investigation by Professor W. F. M. Goss shows that long and indirect pipe connections materially alter the shape and character of the expansion curves. A correct reducing mechanism, practically free from lost motion, must be used so as to reproduce the actual expansion. The arrangement of having one indicator at each end of the cylinder is always to be preferred with engines of considerable size.

The applications of this method must be made with judgment and care. If large leakage exists, only an approximate solution can be obtained, as certain assumptions, treated in Chap. VII, have to be made. Unfortunately only persons who understand thoroughly the different steps involved will be able to use this new method with satisfaction. This is because of the character of the steps upon which the method rests.

6. RELATION OF ADIABATIC VALUES OF $\underline{n}$ FOR STEAM TO EXPERIMENTALLY DETERMINED VALUES.

The adiabatic relations of $\underline{Q}$ and $\underline{n}$ for the pressures used in the tests are plotted in Fig. 7. The action of the cylinder walls is apparently such that no definite relation exists between the two sets

of values. In a general way, it can be seen that the value of Q in a cylinder has a much greater effect on the resulting value of n than for adiabatic expansion.

Owing to lack of time, no extensive study of this subject was undertaken. This relation is not considered to be of much importance.

7. WHY IS THE LOGARITHMIC DIAGRAM MORE USEFUL FOR ANALYSIS THAN ANY OTHER DIAGRAM?

The logarithmic diagram is more useful for analysis than any other form of diagram because of the natural limitations of the human mind. We do not possess the power to distinguish between curves. We are, however, able to see clearly the differences in these curves after they have been transformed into straight lines, which fact alone makes this new method of analysis possible. We are now enabled, in their straight line form, to comprehend curves which we have always seen, but could not distinguish one from another in their original condition.

CHAPTER VII.

APPLICATIONS OF THE NEW METHOD.

1. DETERMINATION OF CLEARANCE.

It was desired to determine the clearance of the diagram shown in Fig. 31. From general knowledge of this class of engines, a trial was made with the clearance assumed as 12.4%, a value purposely assumed as being too small. This value is seen, by the bending of both curves to the left, to be much too small. Trials were therefore made with the clearance assumed as 13.8, 15.1, and 16.0 per cent of the piston displacement. The value of 15.1% gave practically perfect straight lines for both the expansion and compression curves, while the value of 16.0% shows that the lines have begun to bend to the right, indicating too large a clearance. By inspection, it will be seen that the region of fairly straight lines may be located as lying between the values of about 14.5% to 15.5%. The clearance is therefore determined as 15.0%, a value which may be high or low by not more than 4% in this case. This value, 15.0%, is a common value for engines of this class.

This method is more accurate for large clearances than for small ones. The closeness of location of the straight line region, lying between the two families of diverging curves, will be found to be within 5% to 10% of the clearance volume, for values of clearance between about 18% and 2%, respectively.

2. DETECTION OF LEAKAGE.

Very smooth curves may be obtained in the PV-diagram even if there is large leakage taking place. This is seen by referring to Fig. 26, both the expansion and compression curves being fairly regular. The logarithmic diagram, however, shows clearly, in connection

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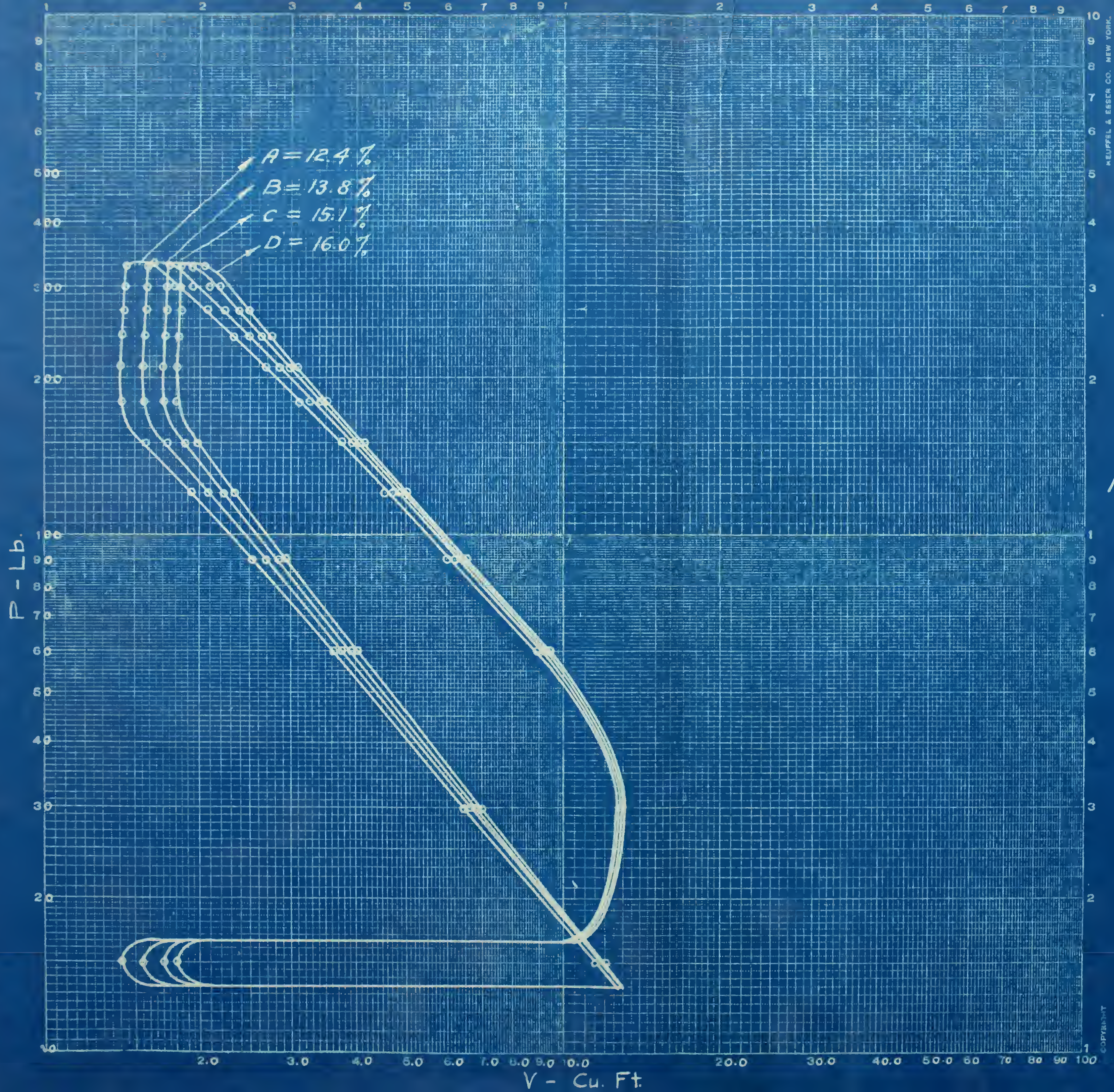

Fig. 31
Koerting Form Cycle
$25\frac{5}{8} \times 37\frac{5}{8}$
Producer Gas

with the discussion and the examples shown, that large leakage of two kinds was taking place during expansion and compression. Leakage which occurs during admission or during exhaust has no effect upon the lines of the diagram, as the weight of the steam present is continually changing.

a. Location of the Various Leaks.

It is seldom found, when leakage occurs in a cylinder, that only one source of leakage exists. Leakage is usually the result of wear, which affects most of the possible sources of leakage in about an equal proportion. As a result, several leaks are generally affecting the curves. This is the case in Figs. 26 and 27. In Fig. 26, leakage of steam was taking place both into and out from the cylinder.

In discussing leakage, it must be kept in mind that difference in pressure is the cause of this phenomenon. In the steam engine there are three pressures which must be considered: the pressure in the steam chest, in the cylinder at the point discussed, and in the exhaust passages. Leakage, being due to difference of pressure, becomes material only when this difference becomes considerable. Thus leakage into or out from a steam cylinder has been found to occur, in most cases, only when the pressure difference is over 30 lb. In Fig. 26, the leakage into the cylinder, shown by the lower parts of the lines, begins to occur at about 25 lb. absolute, or 35 lb. lower than the pressure at admission. The leakage out from the cylinder, shown by the upper parts of the lines, ceases to occur at a pressure of about 40 lb. absolute for the expansion curve, and begins to occur at about 25 lb. absolute for the compression curve. The difference of pressure between the steam in the cylinder and that in the exhaust passage is about 35 lb. in the first case, and about 20 lb. in the second case.

This fact, found on many diagrams analyzed, enables us to

divide the lines roughly into three equal parts on the logarithmic diagram, (when these lines extend from the initial pressure to nearly the back pressure): 1st, the upper third, influenced by leaks out from the cylinder; 2nd, the middle third, practically uninfluenced by leakage; 3rd, the lower third, influenced by leakage into the cylinder. Thus fairly reliable values of n, free from the effect of leakage, may be obtained from the middle third.

Returning now to Fig. 26, both the lines indicate leaks out from the cylinder. This can occur either past the piston or through the exhaust valves. The piston generally becomes leaky sooner than Corliss exhaust valves, and, in this particular engine, one of the piston rings was found to be broken upon examination. When diagrams from both ends of the cylinder are available, piston leakage causes nearly equal effect on the expansion curves of each end. The leakage into the cylinder can come from only one source which can influence the curves, that is, the steam chest. The effect of this leak is seen in both lines in the lower thirds.

In Fig. 27, both forms of leakage are shown in the compression curves for ammonia. The lower thirds show leakage into the cylinder either through the discharge valves or past the piston. The upper thirds of the lines show large leakage from the cylinder, caused by the condition of either the suction valves or the piston. Ordinarily it is not possible to distinguish between two leaks occurring in the same third of the curves.

b. Computation of Leakage.

Fig. 25 is an example where only one kind of leakage is present. Here, to the writer's knowledge, the piston alone leaked badly at the commencement of the stroke. The effect of this leak is seen in the upper third of both lines.

When only one kind of leakage exists it is possible to com-

pute with fair accuracy the leakage taking place during expansion or compression. The lines are extended, as shown in Fig. 25, giving the lines of constant weights of the medium. The volume of gas that had leaked during compression, up to 100 lb. absolute pressure, is then seen to be 0.014 cu. ft., or 6.3% of the volume remaining. The volume of gas measured at 450 lb., that leaked after combustion during expansion, is seen to be 0.032 cu. ft., or 18.7% of the volume remaining after the leakage stopped.

The leakage that took place during combustion at the end of the stroke cannot be computed, but it can be estimated by making the assumption that this leakage was proportional to the mean rate of leakage shown by the two curves, and that its duration was the time interval from the point A to the point B.

The important result that is attained by this method is not, however, the computation of leakage, but the knowledge that it is taking place, so that it can be located and stopped.

c. The Use of the New Method in Testing for Maximum Economy.

Many engines are sold, and their prices fixed, on the basis of their test performance. The importance to the manufacturer of being able to eliminate leaks during this test does not have to be emphasized. Does the engineer in charge of the test know whether the engine is tight?

All our present knowledge of leakage is an inference drawn from the leakage "standing". Nobody knows whether an engine that is tight "standing" leaks when in operation, or vice versa.

This new method gives the only information ever obtained upon this question, for it reveals leakage when the engine is in regular operation.

This method should be applied to all engines about to undergo any test where maximum economy is the object desired.

d. Knowledge as to When General Repairs of the Cylinder and Valves are Necessary.

Leaks are caused by wear, poor design, and accidents.

The accidents include scoring of cylinders and valves, cutting of valves, and cracks in the cylinder.

Most leakage is the result of wear and tear due to long and hard use. After the wear and tear has become marked, it is the custom to rebore the cylinder and to resurface the valves and valve seats.

Several methods are in use for determining when general repairs are necessary for steam cylinders. One of these is to judge the time from the general appearance of the parts on inspection. Cylinders are rebored by some engineers when they have worn "out of round" by a given amount. In small plants the most general method seems to be to wait until the leakage is so large as to become clearly noticeable either by the reduced capacity of the unit, or by the effect upon the coal pile. Some railroad companies overhaul the cylinders and valves at regular intervals of, for instance, 150,000 miles of travel. Some cylinders in stationary plants are rebored at equal time intervals of some four or five years each.

As a pure question of economy, general repairs should take place when the extra annual cost of fuel and water due to leakage equals the annual interest on the money necessary for general repairs. This cost is composed of several items: the actual cost of the repairs, the cost of losing the unit from service, the interest on the cost of the extra capacity that may have to be installed to take the load when units are out of service, and the interest on the money invested in the unit out of service.

It will be appreciated that the existing methods of determining when general repairs are necessary are in a very haphazard

state as regards economy, and that the personal judgment of the man making the decision may be liable to great variation.

The method of detecting leakage from the indicator diagrams offers a rational solution of this important question.

e. Interpretation of the Logarithmic Diagram.

In the discussions upon the effect of wrong clearance, leakage, and excessive condensation upon the lines of the logarithmic diagram, it was assumed, for the sake of clearness, that only one of these effects existed at one time. The examples were selected so as to illustrate only one of these effects in each case.

Cases occur where several of these effects exist at one time in the same diagram. The separation of one effect from another is not an exact process. However, the character of the curves showing excessive condensation, wrong clearance, and leakage are quite different. For instance, wrong clearance affects the lines throughout their length. Excessive condensation, in the cases of the steam diagrams examined, always affects only the upper parts of the curves. Leakage, as has been mentioned, affects only the upper and lower thirds of the lines, where these lines extend from the initial pressure to nearly the back pressure. When excessive condensation and large leakage exist together, no close approximation of the clearance can be made.

An adequate treatment of the segregation of these various effects, when found together, is beyond the scope of this thesis. The treatment is long and complicated. It has been found, however, that long experience in the use of logarithmic diagrams enables one to separate these effects qualitatively.

f. Do Most Engines Leak When in Operation?

After the analysis of about 300 diagrams of a great many engines of all classes, the writer is of the opinion that the great majority of engines, in good condition, are practically tight as regards

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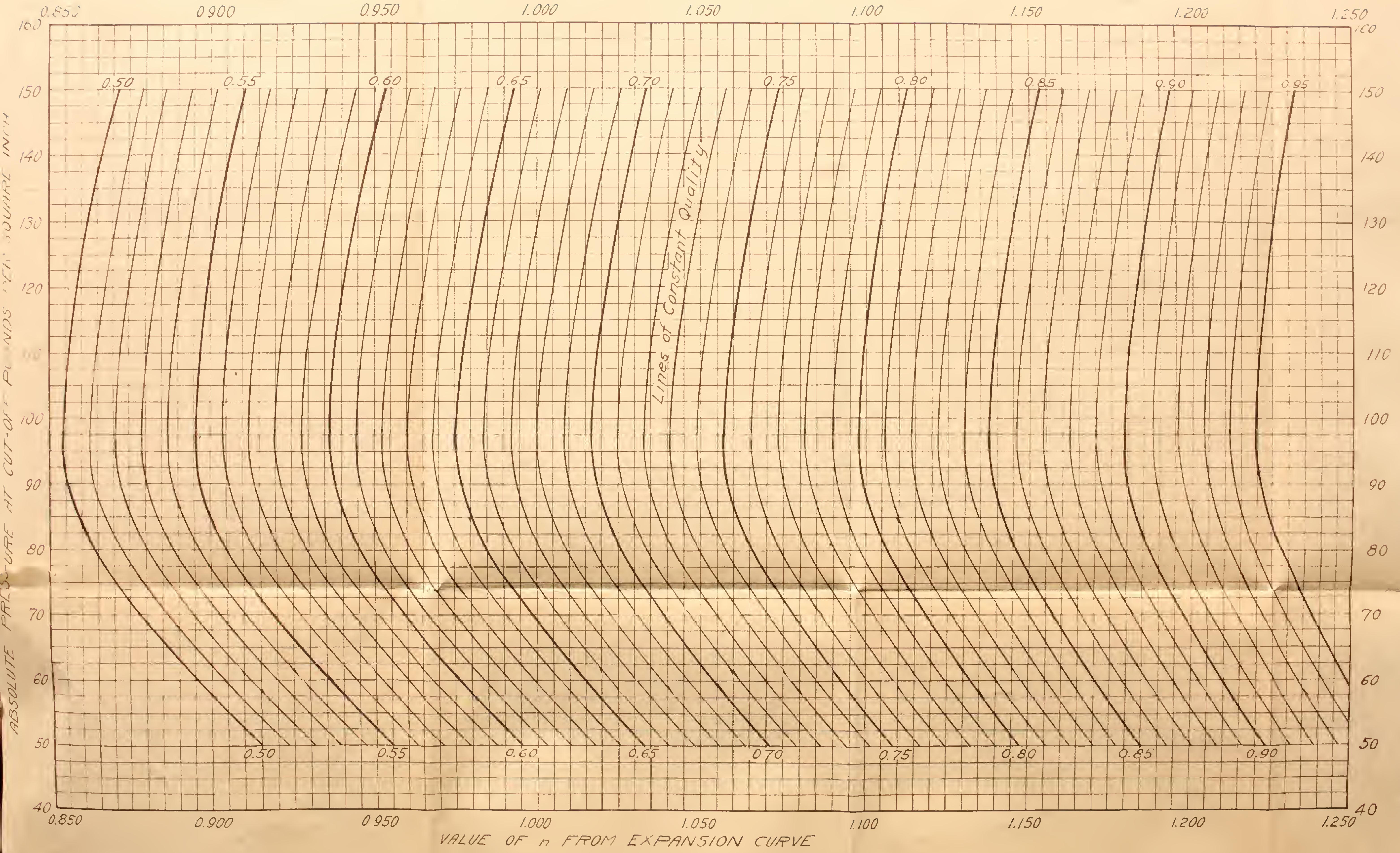


CHART SHOWING THE RELATION BETWEEN PRESSURE AT CUT-OFF
AND VALUES OF n FOR CONSTANT QUALITY AT CUT-OFF

leakage into or out from the cylinder.

3. COMPUTATION OF THE STEAM CONSUMPTION FROM INDICATOR DIAGRAMS.

The relations of Q and n, as determined for various pressures at constant speed from the engine described in Appendix I, were plotted in the form of the chart shown in Fig. 30.

The next step was to examine, with certain restrictions, the tests of other engines, and compare the relations of Q and n with those given in Fig. 30. The restricting conditions imposed were: 1st, that the tests should come from reliable sources; 2nd, that the data supplied should be complete enough to be able to compute the quantities needed for comparison; 3rd, that the cylinders be unjacketed; 4th, that the diagrams furnished should be representative of average conditions; 5th, that the back pressure in the cylinder examined should be practically atmospheric; 6th, that no large leaks should exist.

The values of Q, n, and P, for the set of diagrams to be examined, were first found. Next, the values of n and P were located in Fig. 30, and the corresponding value of Q found as obtained in the tests given in Chap. V. The value of Q obtained from the chart and that obtained from the test examined were compared, and the steam consumption, as computed by the value of Q taken from the chart, was obtained.

The results of ten tests which fulfilled the conditions imposed are given in Table 29. Four distinct classes of engines were examined. These include simple Corliss or Four Valve types, the H.P. cylinders of compound engines, high speed, and locomotive engines. The sizes range from 14 1/2 in. X 13 in. to 28 1/2 in. X 59 1/2 in. and the speeds from 253.4 to 64.7 R.P.M.

The final results, given in Column 26, show that the average steam consumption of all the engines, computed from the diagram, was 2.65% higher than the test results. The average difference between

29

TABLE 29 - COMPUTATION OF STEAM CONSUMPTION FROM INDICATOR DIAGRAMS OF VARIOUS TYPES OF ENGINES

Number of Analysis 1	Class of Engine 2	Maker 3	Where located 4	By whom Tested 5	Date of Test 6	Where published or by whom furnished 7	Cylinder Dimensions inches 8	H.P. Piston Rod Size 9	Clearance H.P. Cylinder, percent 10	Actual Weight of Steam thru cylinder examined per hour 11	Actual Steam per I.H.P. hr., Lb 12	Total I.H.P. Developed 13	Revolutions per minute 14
101	Corliss S.C. Non cond	Unknown	Unknown	Geo. H. Barrus	Unknown	Engine Tests #1	2[illegible] x 60	3 1/4	2.5	84[illegible]	27.77	305.2	74.7
102	" " "	"	"	"		2	28 1/2 x 57 1/2	4	3.0	13070	25.80	506.5	64.8
103	" " " "	"	"	" "		" " 7	26 3/16 x 48	3 7/8	3.0	6742	29.03	232.3	64.7
104	Double Valve 2 cyl "	"		" "		" 10b	17 x 24.2	2 3/4	2.0	7952	25.64	310.1	152.9
105	4 valve S.C. "	"	"			" 11	16 5/8 x 32	2 3/4	4.0	1879	37.43	50.2	79.8
			← Simple Corliss Engines - Non-Condensing →										
151	Corliss Cross Compound	Watts Campbell Co	Thos. Oakes & Co. Bloomfield N.J.	J. R. Oakes W.M. White	1-20-03	Makers	20 x 36 x 36	3	2.5	6740	15.00	433.5	75.83
152	Buckeye " "	Buckeye Engine Co	Dayton & Troy El Ry Co Tippecanoe City, Ohio	Sargent & Lundy	5-6-02	"	18 3/4 x 36 x 36	3 1/2	4.3	8675	14.20	634.2	123.3
			← Compound Engines Condensing H.P. Diagrams Analyzed →										
401	S.C. High Speed Piston Valve	A. L. Ide & Sons	Unknown	Unknown	Unknown	Makers	16 x 16	2 5/8	13.15	4820	26.35	184.6	253.4
402	" Single "	Unknown	"	Geo. H. Barrus	"	Engine Tests #13	14 1/2 x 13	2 1/4	10.0	1740	32.67	53.26	246.0
			← High Speed Engines - Non Condensing →										
451	2 Cyl Simple Locomotive	Schenectady Loco Wks	Purdue Lab Lafayette Ind	W. F. M. Goss	7-3-05	Carnegie Institution Report High Steam Pressure 2[illegible]	16 x 24	2 3/4	H C R 7.44 7.98 L 7.34 7.63	7180	28.40	252.79	97.08
			← Simple Locomotives →										

CONTINUED FROM ABOVE

Number of Analysis 15	Revolutions per hour 16	Weight of Steam Passing thru engine per revolution, Lb 17	Lb of Steam per revolution shown by Indicator at Cut-off 18	Lb of compression Steam saved per revolution by Indicator 19	Lb of Steam Mixture Actually present per revolution 20	Quality of Steam Mixture at Cut-off 21	Absolute Pressure [illegible] Lb 22	Average value of n from expansion curve 23	Quality of Steam Mixture at Cut-off from Chart 24	Weight of Steam per I.H.P. hr. by New method, Lb 25	Percent Steam consumption by New Method is higher + or lower − than measurements by Test 26		By approximate Method: Quality of Steam Mixture at Cut-off from Q=1.245n−.0576 27	By approximate Method: Weight of Steam per I.H.P. hr. by Approximate Method 28	By approximate Method: Percent Steam consumption by Approximate Method is higher + or lower − than measurements by Test 29	By approximate Method: Method giving closer results to Test, C or A [illegible] 30	Remarks 31
101	4482	1.890	1.666	0.130	2.020	0.825	65	0.98	0.767	29.71	+[illegible]		0.790	28.95	+4.2	A	Diagrams used as reproduced in book, assumed representative
102	3888	3.362	2.822	0.167	3.529	0.801	75	1.108	0.795	26.00	+0.75		0.804	25.70	-0.37	A	
103	3882	1.736	1.402	0.116	1.852	0.757	75	1.051	0.[illegible]32	30.00	+3.3		0.733	30.00	+3.2	A	
104	9174	0.867	0.7290	0.0635	0.9305	0.[illegible]84	70	1.131	0.[illegible]	24.60	-4.1		0.833	24.03	-6.3	C	
105	4788	0.3925	0.2732	0.0607	0.4532	0.603	[illegible]	0.964	0.530	38.50	+2.8		0.624	36.12	-3.5	C	
151	4550	1.481	1.244	0.1012	1.582	0.786	100	1.079	0.777	15.17	+1.1		0.767	15.38	+2.4	C	Diagrams furnished were tracings of originals and were representative
152	7398	1.173	1.023	0.1490	1.322	0.774	120	1.060	0.749	14.66	+3.2		0.744	14.73	+3.9	C	
401	15204	0.3196	0.3060	0.0948	0.4144	0.739	110	1.027	0.710	27.[illegible]	+3.9		0.703	27.60	+4.9	C	Diagrams, tracings said to be representative
402	14760	0.1179	0.1197	0.0613	0.1792	0.668	85	0.970	0.637	34.19	+4.6		0.631	34.47	+5.5	C	Diagrams as in book assumed "
451	58248	1.232	1.064	0.1947	1.4267	0.711	100	[illegible]	0.683	29.50	+3.9		0.673	29.95	+5.3	C	Diagrams, originals furnished as representative

the computed and the test results, irrespective of sign, is 3.47%. The maximum differences are 7.0% higher, and 4.1% lower, than the test results.

The results from the approximate method, given in Column 29, show an average difference, irrespective of sign, of 3.96%. These results are not as close as those obtained by the use of the chart.

The results of the applications made up to the present, with the restricting conditions imposed, show that the steam consumption of engines may be computed from the indicator diagram to within an average difference of less than 4% from the test results. Individual examples, however, may show as much as 8% difference in rare cases.

4. FIELDS OF RESEARCH OPENED BY THE NEW METHOD.

a. Study of Leakage through Valves in Single Valve Engines.

It is well known that single-valve engines, as a class, use much more steam than Corliss valve engines under the same conditions. The most important reason for this fact seems to be leakage of steam from the steam chest to the exhaust passage, under the face of the valve, without this leakage steam having passed through the cylinder at all.

Actual measurements of this leakage have been made in tests by Callender and Nicolson, a Committee of the Institution of Mechanical Engineers, and by Mr. George Mitchell of the University of Pennsylvania. All of these experimenters, on account of the nature of the tests, were forced to operate the engine tested under abnormal conditions.

The new method of accounting for the actual weight of steam mixture present in the cylinder at cut-off, offers a very useful tool for research in this field. The engine may be operated under absolutely normal conditions while being experimented upon.

b. Combined Logarithmic Diagrams of Multi-expansion Engines.

The study of the combined logarithmic diagrams of multi-expansion engines is very profitable in the light of the new method of analysis. A combined diagram from a triple-expansion pumping engine, 28 in. X 54 in. X 80 in. X 60 in., is given in Fig. 32.

The weight of steam mixture present may be accounted for from cylinder to cylinder by means of the relations of Q and n for this class of engine. This method of accounting would be very useful in investigating the problems of reheating, steam jackets, and the choice of cylinder ratios.

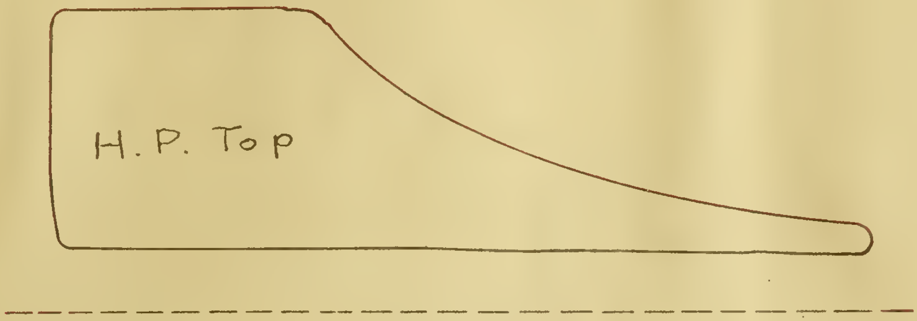
c. Comparing the Leakage of Engines "Standing" and in Operation.

The standard methods of testing engines for leakage are only applicable when the engines are at rest. Callender and Nicolson, in their leakage experiments, observed that while a slide valve may be perfectly tight at rest, yet this same valve may leak enormously in motion, due to the breaking up of the oil film.

A comparative study of the leakage of engines, both at rest and in motion, by the new leakage method would be a very fruitful field of research.

d. Study of the Springing of Locomotive Valve Gears.

The valve gears of locomotives, due to the spring caused by the inertia and lost motion of the parts, give a later cut-off at high speed than at low speed with the reverse lever in the same notch. Various expedients have been used to measure the amount of this spring. The most accurate method in use at present is to take valve ellipse diagrams from the valve rod. Experiments using this method have been undertaken by the University of Illinois. The Pennsylvania R. R. has attempted to measure the amount of this spring from the difference of cut-off between high and low speeds with the same reverse lever position as shown by the PV-diagrams.



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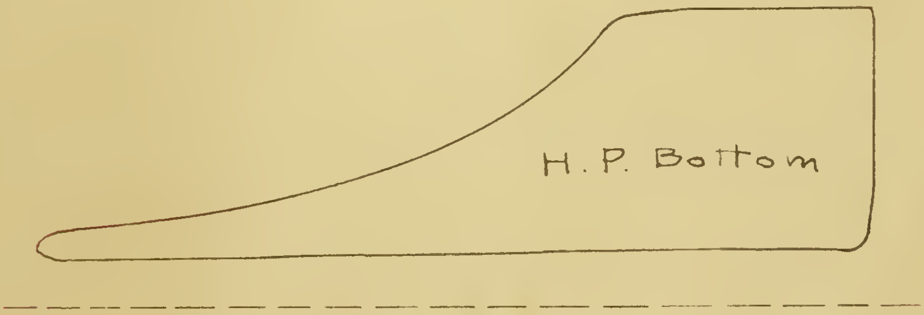

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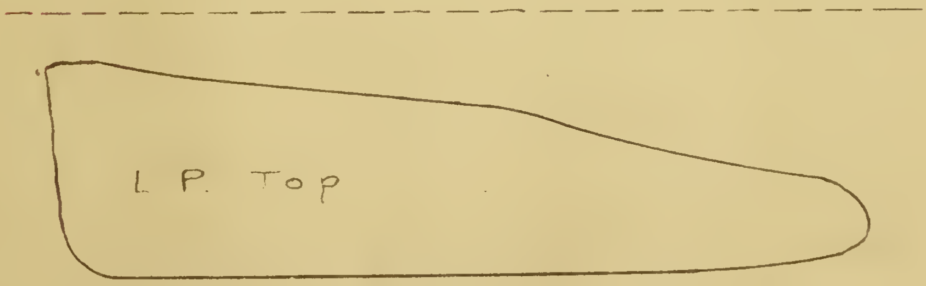

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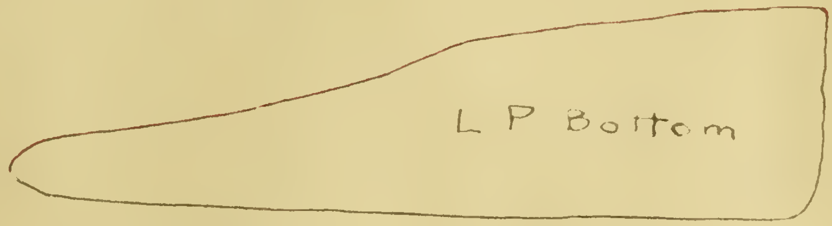
Fig. 32



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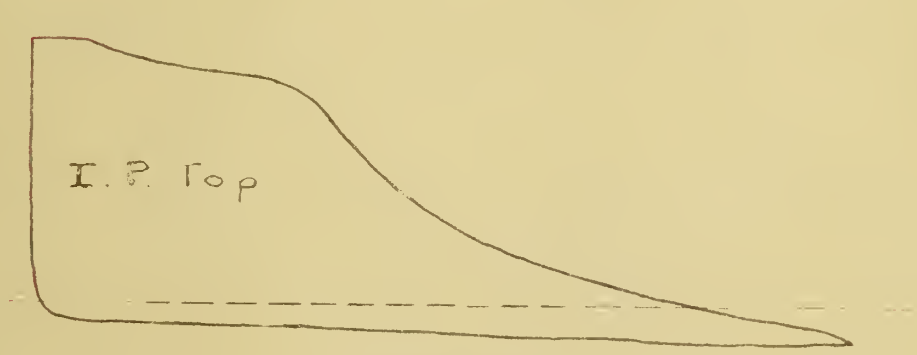

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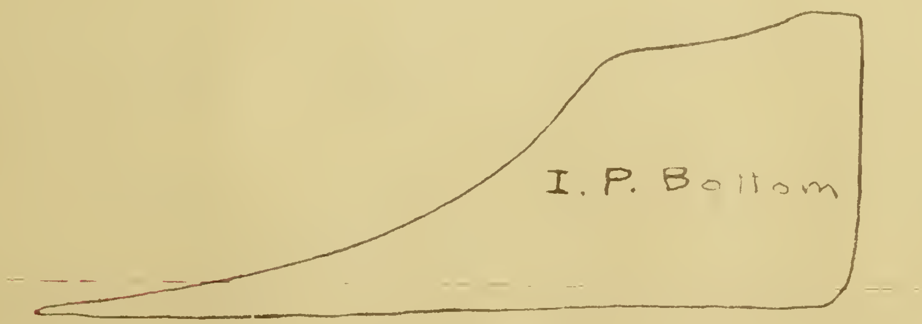


Scale 20#
Fig. 32

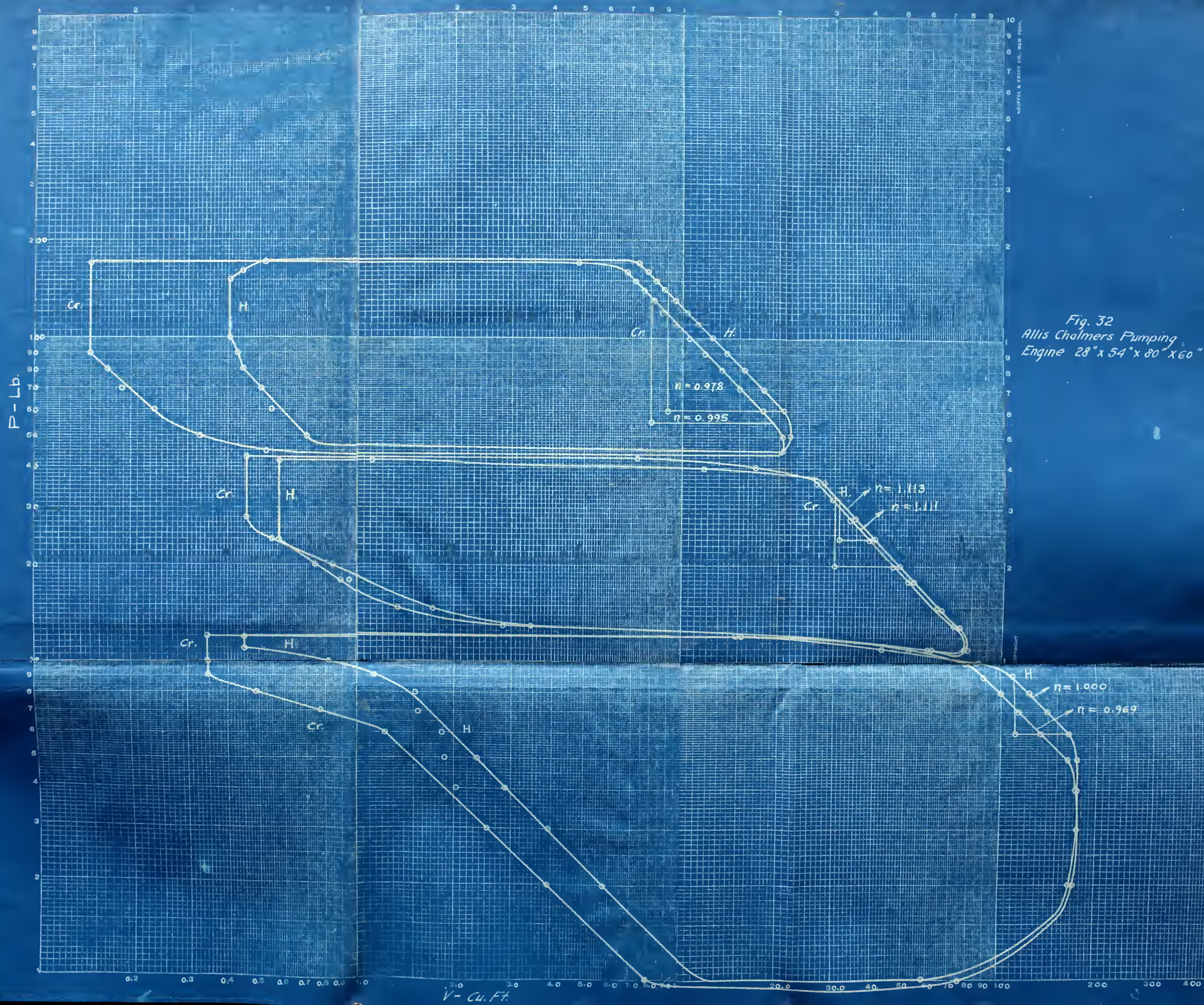


Fig. 32
Allis Chalmers Pumping Engine 28" x 54" x 80" x 60"

The extremely close location of cut-off obtainable from the logarithmic diagram enables the study of the spring of valve gears to be carried on with only the diagrams taken at various speeds with the same reverse lever position.

CHAPTER VIII.

COMMON ERRORS MADE IN ANALYZING STEAM DIAGRAMS BY EXISTING METHODS.

1. THE USE OF THE EQUILATERAL HYPERBOLA AS A STANDARD OF COMPARISON FOR STEAM DIAGRAMS.

The values of n for the expansion curves of steam PV-diagrams are subject to very wide variation. The extent of this variation may be seen from the examples given in Chap. III. The range of variation in the examples analyzed in the present investigation is from about 0.70 to 1.35. The range of values in the engine tested was from 0.835 to 1.234. It is true, however, that the average of all values obtained is close to 1.0. The average value for the engine tested is 1.004.

The values of n for most engines of ordinary size using saturated steam at normal cut-off are not far from 1.0. The explanation of this value can be easily seen from the results of the tests given in Chap. V, and in the applications given in Chap. VII. The only meaning whatever that this average value of n = 1.0 ever possessed, is that the average value of Q, in the class of engines examined, lies in the range between 0.60 and 0.70.

The explanations given by various writers for the average value, n = 1.0, are nothing short of ridiculous when the facts are known. The actual value in any particular case has been represented by some writers as being the erratic result of a hopeless number of variables. Other writers have discovered to their satisfaction that the equilateral hyperbola represents the only true theoretical expansion in a cylinder, and call such expansion Mariotte's law. The law of Boyle or Mariotte, or the law of isothermal expansion of a perfect gas, has no more bearing on the expansion of steam in a cylinder than

has the moon. This last class of writers, when finding a case where n is much higher or lower than 1.0, regard such a case as the result of abnormal and mysterious conditions, and they generally end up by calling these conditions leakage.

Some writers have noted that superheated steam gives higher values of n than 1.0. Did the value of n rise to this higher value at one jump?

The only rational use of applying the equilateral hyperbola to steam PV-diagrams is to act as guide to see whether n is greater or less than 1.0. If the actual line is not close to this hyperbola, then this fact means that the value of Q for the case examined is less than about 0.60, or greater than about 0.70. The assumption that $n = 1.0$ as a standard of expansion is equivalent to assuming that the value of Q is standard at about 0.65.

The elaborate theory of analysis built on the assumption that $n = 1.0$ is the natural result in any art where the actual facts have never been investigated.

The use of the equilateral hyperbola to predict the form of PV-diagrams for the purposes of design is entirely warranted in the case of ordinary sizes of engines using saturated steam.

When steam superheated over 100°F is used, the value of n should be assumed at between 1.10 and 1.20. The high values of n obtained with highly superheated steam in large engines alter materially the division of work and the tangential forces acting from those obtained when n is assumed to be 1.0. This fact should be considered in the design of engines to use superheated steam.

The use of the equilateral hyperbola to obtain the ratio known as the "diagram factor" has no rational basis, but its use for this purpose gives results which are valuable for the purposes of design.

2. THE METHOD OF APPROXIMATING THE CLEARANCE OF STEAM CYLINDERS BY ASSUMING n = 1.0.

If n has the value 1.0 on a PV-diagram, the clearance may be found by locating the zero of volume on the zero line of pressure. This process is done graphically by reversing the method used in constructing the equilateral hyperbola.

In actual expansions, however, n is almost never exactly equal to 1.0, but is greater or less as already explained. The accuracy of the result by this method is dependent on how close the value of n approximated 1.0. The clearance obtained by this method may be as much as 100% larger or smaller than the actual clearance volume in ordinary cases, while errors of 25% and 50% are very common.

A rational method of approximating the clearance cannot be based upon the assumption that n = 1.0, but only upon the fact that it be of a constant value, the value itself being immaterial.

3. DIVISION OF FEED FOR APPLYING HIRN'S ANALYSIS.

One of the requirements of Hirn's analysis is that we know exactly how much steam was admitted to each end of a cylinder. These amounts are not usually equal in practice, so an assumption must be made to cover the needs of the case.

The usual assumption is to divide the feed between each end of a cylinder in the ratio of the values of the mean effective pressure shown by the PV-diagrams from the two ends. This assumption is probably not far from the actual division in most cases, and is the best that can be done under the circumstances.

In the light of the facts presented in this thesis, this feed may now be divided on a more rational basis. It has been found that the presence of the piston rod in only one end of the cylinder has no appreciable effect upon the value of n. This fact enables us

to divide the feed according to the volumes filled and the values of Q as determined by the resulting values of n. This method is believed to be the closest solution obtainable in the case where the supply for each end of the cylinder cannot be separately determined.

4. COMBINED STEAM DIAGRAMS.

Very little of value is obtained from the combined PV-diagrams of steam engines, except the measure of the diagram factor for the purposes of design.

One of the uses that has been made of the combined diagram is to see whether continuity of expansion exists. It has been assumed by various writers that continuity of expansion should exist.

A study of the relations of Q and n shows that continuity of expansion does not and should not exist except under very special conditions.

5. COMPUTING THE WEIGHT OF STEAM RETAINED IN COMPRESSION.

The best obtainable investigations indicate, when saturated steam is used, that the steam in the cylinder is dry at the closure of the exhaust valve. As shown by the logarithmic diagrams, leakage of the steam in compression continues until the exhaust valve, in closing, has acquired considerable seal.

The point,which is selected to compute the volume of steam retained in compression generally lies between the points A to B, and C to D of Fig. 1. As the weight of steam retained is yet decreasing, the point selected nearly always accounts for more steam than was actually retained. In other words, we do not find the value of Q to be as high as it actually is.

The following method, shown in Fig. 1, has been adopted in the present investigation. The straight line of the compression curve on the logarithmic diagram, or the line of constant weight of steam mixture, is prolonged dotted as shown to the back pressure. The in-

tersection of this prolonged line with the back pressure line extended is taken as the volume of dry steam retained in compression.

This method almost always gives less steam retained in compression than the ordinary method, and is believed to be rational in the present state of knowledge of this subject.

CHAPTER IX.

CONCLUSIONS.

The following conclusions have been drawn from the data collected in this research, as applicable to reciprocating engines using elastic media, and having as a part of the cycle of operation, an expansion, a compression, or both.

1. The indicator diagram, taken with a correct reducing motion and with a reliable indicator, contains the evidence necessary for a very complete and useful analysis of cylinder performance.

2. The logarithmic diagram, derived from the indicator diagram, discloses a new and very complete analysis of cylinder performance.

3. Leakage(if appreciable)may be reliably detected from the logarithmic diagram, and may in some cases be computed in volume.

4. The clearance of cylinders may be found by graphical trial on logarithmic paper to within 5% to 10% of its actual value, depending as the clearance itself varies from 18% to 2%, respectively, of the piston displacement.

5. The cyclic events, even though entirely obscure in the indicator diagram, may be located on the logarithmic diagram to well within 1/16 inch, this quantity being the equivalent of about 1/25 inch when retransferred to the indicator diagram.

6. The actual water consumption of steam engines may be obtained by means of a new method from the indicator diagram, within the limitations imposed, to well within an average of 5% of the water consumption as measured by test.

7. This method of computing steam consumption is more accurate than the average test, and is the only accurate method available

for testing certain classes of engines.

8. This method of computing the actual steam consumption has one important advantage over time tests: it virtually measures a rate instead of a quantity.

9. The various phases of this new analysis can be satisfactorily applied only by persons who comprehend the various steps involved, and who by extensive experience, possess broad judgment.

APPENDIX I.

DESCRIPTION OF THE PLANT.

1. APPRATUS USED.

The engine used for the tests given in Chap. V is of the Corliss type, and is a part of the equipment of the Mechanical Engineering Laboratory at the University of Illinois. The plant comprises the engine and various auxiliaries which are used in connection with the engine. These auxiliaries consist of a throttle valve in the steam pipe line, an independently fired superheater, a direct current generator with a water rheostat, a surface condenser, scales, and a tank. The instruments used consisted of indicators, pressure gages, thermometers, ammeters, voltmeters, and a continuous revolution counter.

The general arrangement of the plant, with the exception of the superheater and the steam piping, was the same for both saturated and superheated steam tests. Steam was obtained from the main boiler plant located about 150 feet away from the engine. Two pipe lines, one of 6 inch pipe for saturated steam and the other of 4 in ch pipe for superheated steam, traverse the laboratory and each is connected to the steam pipe of the engine tested. For the saturated steam tests, the steam was passed at boiler pressure through the 6 inch line from the boiler room for 90 feet until it entered the laboratory; after the line entered the laboratory, the steam was throttled by means of a gate valve in the line to the desired pressure, as shown by a gage on the pipe line on the engine side of the throttle valve; the steam, now at the desired pressure, then passes through 60 feet of 6 inch pipe and 8 feet of 4 inch pipe to the engine. For the superheated steam tests, the steam used was passed as before to the laboratory through

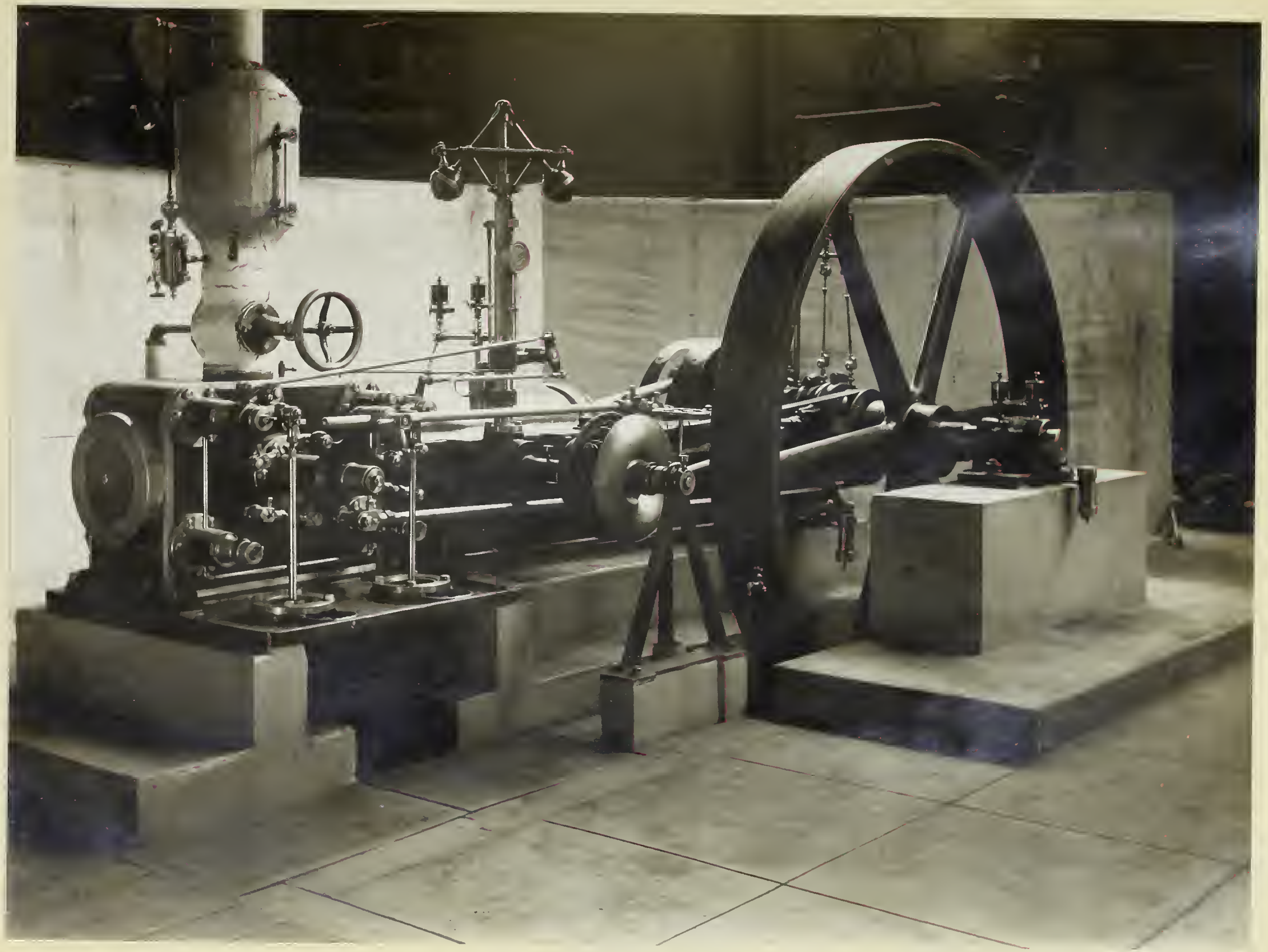


Fig. 33.

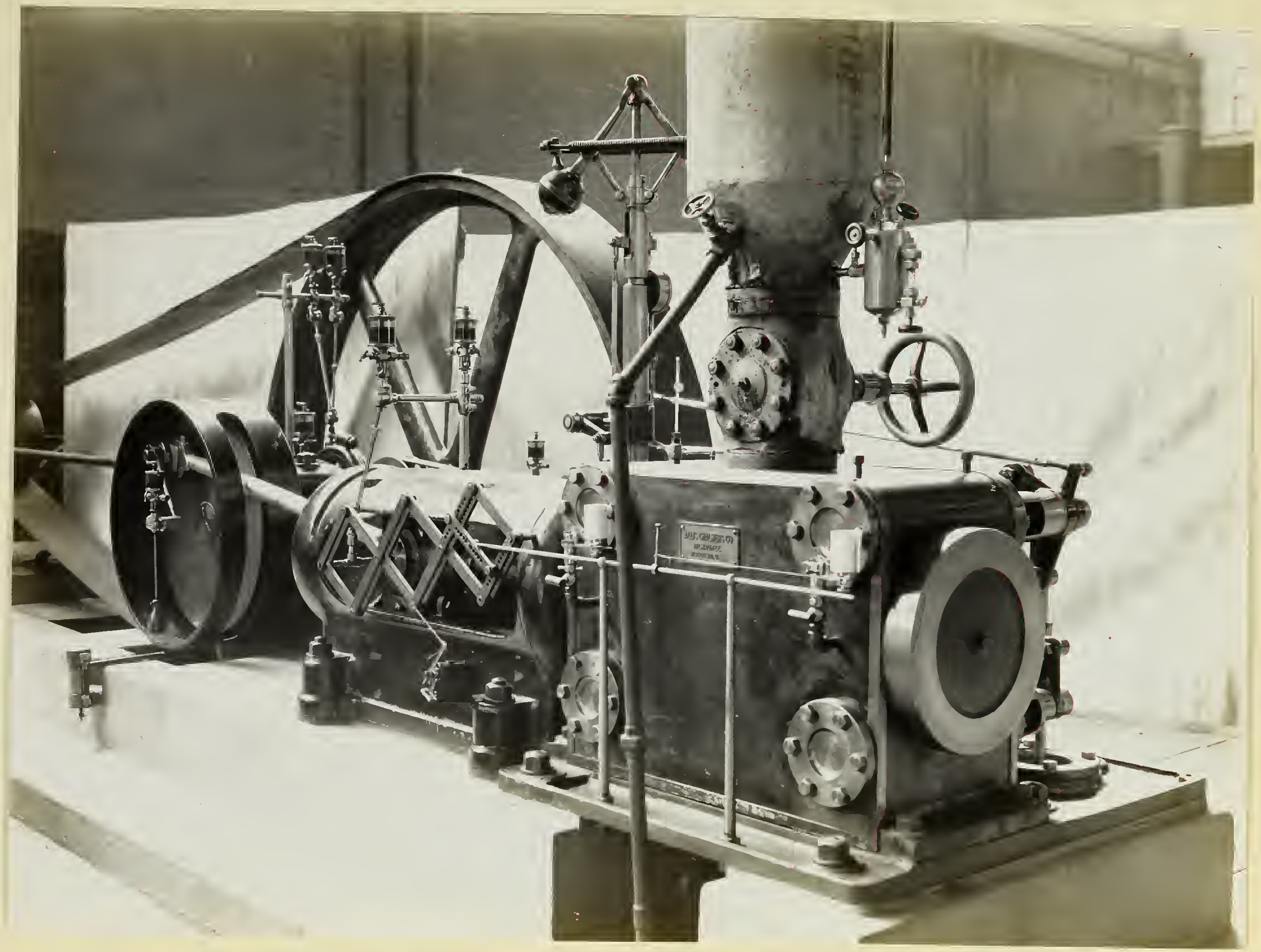


Fig. 34.

the 90 feet of 6 inch pipe; from here it was passed through 20 feet of 3 inch pipe and 10 feet of 3 inch pipe where a gage and a gate valve were located and were used to throttle the steam to the pressure desired; after this valve the steam passed through 66 feet of 3 inch pipe to the superheater, then through 67 feet of 3 inch pipe to the laboratory 4 inch superheated steam main; then through this 4 inch main 61 feet to the engine.

The steam exhausts from the engine through 30 feet of 5 inch pipe to the condenser, where the exhaust steam is condensed, and is then weighed in a large tank on a platform scales.

The engine is connected by a belt to the generator which furnishes the load. This generator is loaded upon a water rheostat located close by.

a. The engine.

The engine is a small well-designed Corliss engine of a standard type built for heavy duty service. Its principal dimensions are given in Table 30. A view of the right side of the engine is shown in Fig. 33, and a view of the left side is given in Fig. 34.

The cylinder is unjacketed on the ends, but is partly jacketed on the barrel by the steam chest, the latter covering about one-fifth of the barrel surface. The exhaust passages are separated from the lower part of the cylinder barrel by a dead air space formed in the cylinder casting.

The engine is fitted with separate eccentrics for actuating the exhaust and the steam valves, thus enabling the steam valve gear to cut-off up to about 50% of the length of the stroke. The two separate eccentrics and the two wrist plates are well shown in Fig. 33.

The speed of the engine is controlled by a fly-ball governor acting on the cut-off cams of the steam valve gear. Variable speed is obtained by varying the gear ratio between the main shaft and the gov-

1-04-1000-R

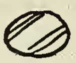
Table 30

Mechanical Engineering Department

UNIVERSITY OF ILLINOIS

Name ...... Date 4/11/10

Report of some observations and measurements of a steam engine at M. E. Lab., University of Illinois, Urbana, Ill.

1. Engine made by Allis Chalmers Co., Milwaukee
2. Usual r. p. m. is 120, variable 20-160.
3. Usual boiler pressure 130
4. Diameter of cylinder 12.02" (hot) Clearance {H - 7.89%, Cr - 7.04%}
5. Length of stroke 24"
6. Length of conn. rod is 5' 6" Its form of cross-section at center 
7. Ratio of crank to conn. rod is 1 to $5\frac{1}{2}$
8. Area of bearing surface of crosshead on guides is 78 square inches.
9. Diameter of the piston rod is $2\frac{3}{16}$ inches.
10. Diameter of shaft in bearings is 6 "
11. Diameter of flywheel bore is 7 "
12. Diameter of belt pulley is 9 feet. Its face is 12 inches.
13. Width of belt is 14 inches. Its thickness $\frac{1}{4}$ inch.
14. Diameter of flywheel is 9 feet. Cross-sect. of rim 32 □"
15. The weight of the flywheel rim is about 925 lbs.
16. The velocity of the flywheel rim is about 56.5 ft. p. s.
17. Length of crank shaft bearing is $15\frac{1}{2}$ inches. Outboard bearing is ......
18. The flywheel or belt pulley is made in 1 piece. Fig. 1 sketches shows method of connecting segments.
19. The governor makes ...... r. p. m. It is shown in Fig. 2 of sketches with a description of its action in regulating the speed of the engine.
20. The number of valves affecting the steam distribution is 2
21. When there is more than one cylinder, their arrangement is shown in Fig. 3 of sketches.

2. The engine is
   - Horizontal. ~~Vertical.~~
   - Condensing, or Non-condensing.
   - ~~Compoun~~d. Single cylinder.
   - Automatic cut-off. ~~Throttling~~.
   - Running over. ~~Running under~~.
   - ~~Self-contained.~~ Not self-contained.

Corliss Double Eccentric

H.P. Constants

H - 0.00688

Cr - 0.00665

23. Examine arrangements for lubrication and taking up wear.
24. Remarks:

Steam Ports $1\frac{1}{16}'' \times 11\frac{5}{64}''$

ernor. This gear ratio change is accomplished by the mechanism shown to the left of the fly-wheel in Fig. 33. The mechanism is operated as follows: the belt from the main shaft drives the concave disc to the left which is loose on the shaft; this disc drives by friction 3 fibre-rimmed idlers which are mounted between the 2 discs on frames supported by a stationary frame in a manner which permits the plane of the idlers to be changed; the concave disc to the right is friction-driven by the 3 idlers and is keyed to the shaft which is connected through bevel gears to the governor. The discs are kept in contact with the idlers by an end thrust provided by a helical spring(surrounding the shaft)which is located between the right disc and the out-board bearing; the left disc is loose on the shaft but works against a collar formed by the sleeve carrying the shaft; the spring, therefore, acting against the outboard bearing forces the right disc against the idlers and the left disc against its collar. The hand wheel shown in the figure, by a gear and sector device not shown, changes the plane of the idlers with respect to the shaft, and therefore changes the gear ratio between the 2 discs. This device permits any speed from 20 to 160 R.P.M. to be obtained.

A leakage test of all valves and the piston, the engine being at rest, showed that these parts were fairly tight.

b. The superheater.

The superheater is of the Foster separately-fired type, and is rated at 200°F of superheat for a flow of 4500 lb. of steam per hour. The draft is induced by an engine driven fan.

c. The condenser.

The condenser is of the Worthington surface type having 362 sq. ft. of condensing surface. Two pumps are provided; one, a Worthington circulating pump drawing its water from the creek about 40 feet away, the other, a Blake wet air pump, discharging into a tank on a

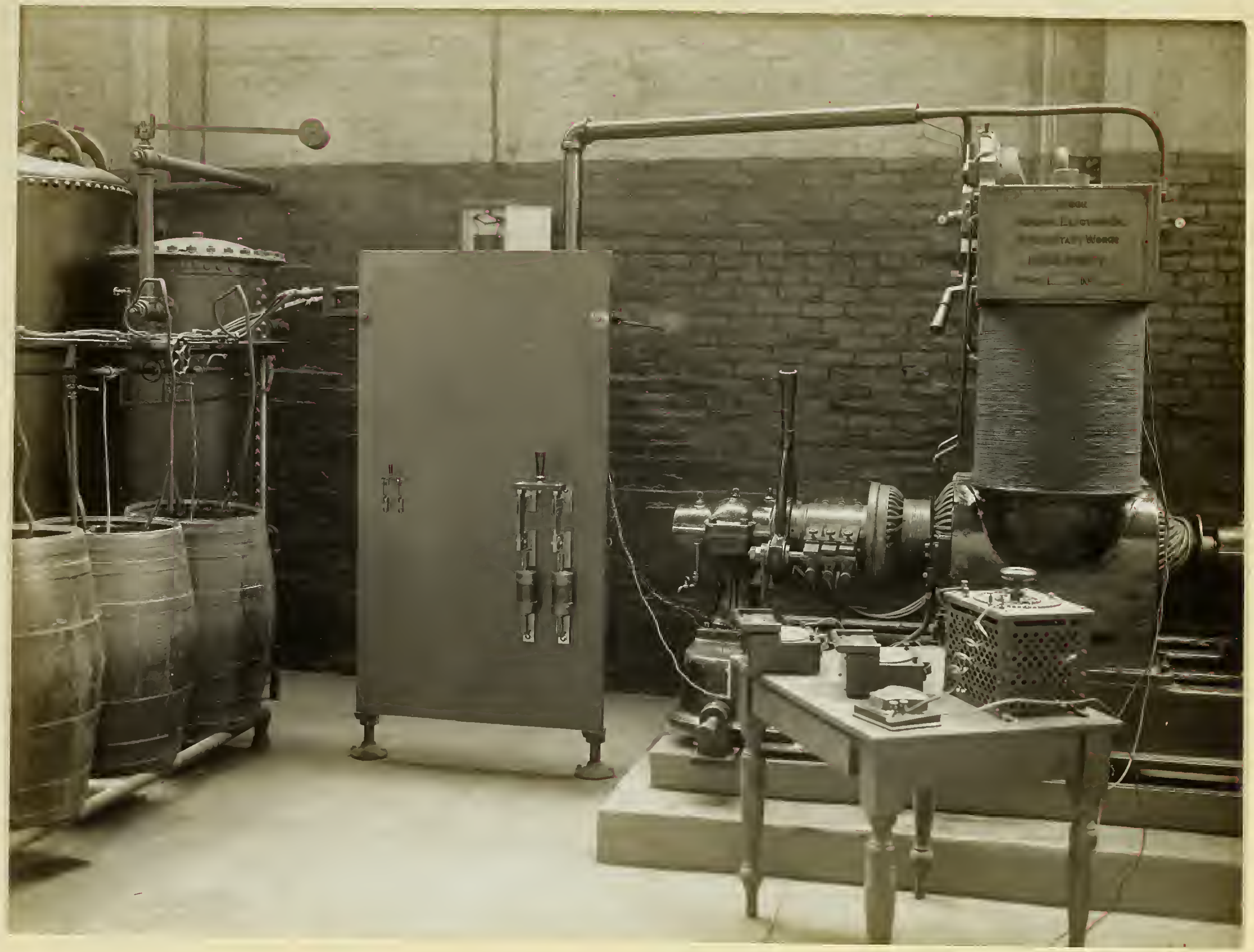


Fig. 35.

platform scales. Tests at frequent intervals showed that the condenser was practically without leakage.

d. The generator and water rheostat.

The engine was loaded on the generator and water rheostat shown in Fig. 35. This generator is of the Edison bi-polar type, and is rated at 100 K.W. at 140 volts, thus giving a full-load current of 715 amperes. The field was separately excited for the tests, the current being obtained from the laboratory 220 volt direct current supply.

Fig. 35 shows the arrangement of the loading part of the plant. The generator output was conducted through cables in the conduit shown to the main switch on the switch-board; here connections were made to the plates of the water rheostat. Voltmeter and ammeter connections are made on the back of the switch-board. The table shown to the right of Fig. 35 contains the voltmeter and milli-voltmeter for the main load current, and the field ammeter and rheostat for controlling the voltage.

The water rheostat consists of 3 water-tight barrels of wood, each containing two iron plates, one positive and one negative. The connections of all the barrels are in parallel. Referring to Fig. 35, and describing only one barrel, a plate connected to the negative terminal is placed on the bottom of the barrel; the positive terminal is connected through the looped wire shown to the front to a movable plate suspended by a rope from the long shaft hung under the frame carrying the conductors. This long shaft is turned by means of the handwheel and worm device shown. The rope holding the upper plate is tied through a hole in the shaft, so that when the shaft is turned the rope is wound up, or vice versa, as desired. The desired load as shown by the milli-voltmeter is obtained by regulating the distance between the plates and by throwing salt in the water to obtain the desired resistance. An overflow from the top of each barrel is provided

at the back, so that when the water boils, cold water is admitted so that the heat is carried away in the overflow water instead of by the boiling of the water. All the upper plates are lowered simultaneously by means of the long shaft. An almost absolutely constant load is maintained by this rheostat.

e. The instruments.

The various instruments used were carefully calibrated.

2. INDICATORS.

The indicators and the indicator rig are shown in position in Fig. 34. Two Crosby inside spring indicators in very good condition were used for the tests. The pipe connections consist of 5 inches of 1/2 inch pipe for each indicator. The reducing motion is a wooden pantagraph mounted as shown. The lost motion in the indicator rig was less than 1/100 inch.

Very great care was observed in the indicator work, since the indicator diagrams themselves formed the basis of the results to be obtained. The pistons were oiled at intervals of about one hour.

The springs were calibrated by steam in the indicators used by a Cooley Fluid Scales Tester, using a method which duplicated almost exactly the conditions under which the indicators were used during the tests.

APPENDIX II.

METHODS.

1. TEST METHODS.

The object of this investigation was to examine the values and relations of the exponent $\underline{n}$ obtained from the expansion curve of the indicator diagrams under different conditions of pressure, speed, and cut-off. The effect of changing one of these variables was studied while keeping the other two constant. The method of testing was planned therefore with a view to maintaining the conditions of pressure, speed, and cut-off as constant as was possible during any one test.

Tests may be very short when a surface condenser is used and constant conditions are maintained. Prior to running a test, the engine was operated for about one hour. It was found that after the operating conditions for one test had been maintained constant for 10 minutes that the steam condensed per unit of time was almost exactly constant. After the conditions had become constant, it was found that 30 minutes of operation gave a length of test which produced trustworthy and consistent results. All observations and diagrams were taken every 3 minutes. Six observers were required to maintain the desired conditions of load, pressure, superheat, and to take the readings.

The steam pressure was closely controlled by an observer who throttled the steam in the main before it reached the engine to the pressure desired as shown by a gage at the valve. Ordinarily this pressure was maintained to within 2 lb. of the pressure desired. It was decided to keep the steam temperature for the superheated steam tests constant at 500° F at the superheater. The superheater was operated by an observer who was guided by the indication of a thermometer

placed in the pipe carrying the steam leaving the superheater. Ordinarily the temperature of the steam was kept within the limits of 490° to 510°F. The temperature variation at the steam chest of the engine did not exceed 2° or 3°F during one test. The load was kept to within about 1% variation of the average load during one test. A back pressure of about 1/2 lb. above the atmosphere was maintained at the engine by keeping the vacuum at the condenser at 1 inch of mercury, this vacuum being controlled by an observer who regulated an air leakage valve on the condenser.

The testing crew became expert in controlling the conditions so that the variations from constant conditions were remarkably small for test work. When the conditions of one test varied for any reason more than the amounts given, this test was not used for the purposes of the investigation.

The tests were designated in the laboratory by a symbol indicating the conditions to be maintained. Thus test 500° - 100 - 52 - 120 indicated that the steam was to be superheated to 500°F at the superheater; that 100 lb. gage pressure was to be maintained at the engine throttle; that 52 K.W. load was to be put on the generator; and that the speed was adjusted to be 120 R.P.M., measured at no-load.

2. METHOD OF SELECTING ONE SET OF INDICATOR DIAGRAMS TO REPRESENT THE AVERAGE CONDITIONS OF ONE TEST.

After a test was run, the constancy of the conditions of pressure, superheat, load, the number of revolutions and the weight of the condensate for each three minute reading was examined. If the variations of these conditions were within the limits selected, the test was worked up in the usual manner. It was generally found that the area of each of the indicator diagrams for each end of the cylinder was within 2% of the mean area. The gage pressure for each reading

at which the diagrams were taken was found to vary less than 2 lb. from the average. To represent the average conditions of one test, the simultaneous combination of a gage pressure reading nearest to the average and of one set of diagrams which had an area nearest to the mean area, was sought. This combination gave one set of diagrams, taken at the average pressure, which represented the average area of all diagrams. This mean combination condition could generally be satisfied to within 1/2 of 1% of the average area.

The value of the average quality of the steam mixture present in the cylinder at cut-off, and the average value of $\underline{n}$ for both expansion curves were obtained from this set of diagrams selected as representing average conditions. The unit of measurement of both quality and $\underline{n}$ was therefore the revolution.

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